NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE

No. 1249

A STUDY OF PISTON-RING FRICTION

By James C. Livengood and Chapin Wallour

Massachusetts Institute of Technology

Washington
August 1947

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE NO. 1249

A STUDY OF PISTON-RING FRICTION

By James C. Livengood and Chapin Wallour

SUMMARY

A series of tests was made with a special engine equipped with a crosshead and an elastically mounted combustion cylinder. The apparatus permitted the isolation and measurement of the friction forces existing between the piston rings and the cylinder wall during operation of the engine. Various combinations of piston-ring and cylinder-sleeve materials were investigated in addition to the effects of several engine operating conditions.

It was found that under the conditions of the tests the use of a porous chromium-plated cylinder caused slightly greater ring friction than a smooth steel cylinder and that a porous chrome-plated top piston ring likewise increased the friction, although to a smaller degree. It was also found that piston-ring friction increased with increased engine speed and with increased manifold pressure, but decreased with increased cylinder jacket temperature.

INTRODUCTION

The investigation described in this report is a continuation of the work begun by Forbes and Taylor (reference 1), who isolated the combined piston-skirt and piston-ring friction in an operating engine by means of a special apparatus. Forbes and Taylor were able to demonstrate that the piston and ring friction increased with increased oil viscosity and increased slightly with increasing indicated mean effective pressure. The combined piston and ring friction was greater at higher values of engine speed. These results were of a preliminary nature, however.

The work reported by Leary and Jovellanos (reference 2) extended the use of the original apparatus to show that piston and ring friction decreased quite rapidly during the first hour of running (starting with new rings and cylinder) and more slowly for an extended period thereafter. An interesting result of this investigation was the observation that the friction work measured during the process of ring-scuffing was not greater than that for normal operation.

The results of these previous investigations appeared to show that the technique developed was sound and could yield useful information. It was decided, therefore, to extend the studies to include additional variables (such as ring and cylinder-barrel materials) and to check the original results of reference 1.

In the interests of isolating the variables as completely as possible, however, it was decided that the apparatus used in the tests reported in references 1 and 2 should be changed by incorporating a crosshead. This modification would permit operation of the engine with only the piston rings in contact with the combustion cylinder.

A further change in the apparatus was the use of much stiffer diaphragm springs for the combustion-cylinder suspension. The object of this change was to increase the natural frequency of the cylinder and thus improve the detail of the friction records obtained. This increase in stiffness of the measuring system required a more sensitive recording device, and consequently an electromagnetic arrangement was substituted for the original optical system.

The foregoing changes in the apparatus were incidental to investigating the effects of engine speed, load, and cylinder temperature on piston-ring friction. This investigation was conducted at the Massachusetts Institute of Technology under the sponsorship and with the financial assistance of the National Advisory Committee for Aeronautics.

The authors are indebted to Mr. W. A. Leary and Mr. J. U. Jovellanos, whose previous experience with the apparatus was invaluable; and to Professors E. S. Taylor and C. F. Taylor for criticism and advice.

DESCRIPTION OF APPARATUS

Engine

Cylinder and cylinder head.- The method used in this investigation was basically the same as the method used by Forbes and Taylor (reference 1) and Leary and Jovellanos (reference 2). It consisted in elastically mounting the combustion-cylinder sleeve so that friction forces between the sleeve and the piston rings could produce a small motion of the sleeve parallel to its axis. The motion was recorded during operation and provided a measurement of the instantaneous friction forces.

Figure 1 shows the cylinder and crosshead assembly. The light cylinder sleeve was clamped on the inner circumferences of two annular steel diaphragm springs. The outer edges of these diaphragms were clamped to a heavy cast-iron cylinder. The space between sleeve and cylinder formed the water jacket. The clamping was accomplished by the cylinder head at one end and a steel plate at the other.

The cylinder head was provided with unsplit junk rings which, together with oil supplied under pressure, formed a seal which effectively closed the combustion chamber against gas leakage. These rings had approximately 0.002 diametral clearance within the sleeve. The junk-ring grooves were deep enough to allow the rings to center themselves properly with the sleeve. The lands between the junk rings had a diameter small enough to ensure that there was no contact with the sleeve.

Vent holes were provided in the cylinder head to allow oil and gases which leaked past the junk rings to escape and thus avoid a rise in pressure above the upper diaphragm. In order to reduce such gas leakage to a minimum, a metered supply of oil was introduced under pressure into a passage leading to the junk-ring grooves. Most of this oil escaped through the leak-off passages above the diaphragm, but some found its way into the combustion chamber and onto the cylinder wall, where it provided piston-ring lubrication. The cylinder head was cooled by cold water flowing through its jacket passages. Low temperature of the head was desired in order to keep the oil viscosity at the junk rings as high as possible.

Two spark-plug wells, which are shown in figure 2, were sealed off from the jacket coolant by rubber seals which exerted no appreciable constraint on axial motion of the sleeve.

<u>Crosshead and piston</u>.- A water-jacketed crosshead cylinder was installed between a CFR crankcase and the cylinder assembly described previously. An aluminum-alloy crosshead, operating in this cylinder, carried on its upper end a special piston. (See figs. 1 and 3.) Since no wrist pin was required for this piston, the central portion was reduced in diameter to decrease weight. The crosshead and piston assembly was designed so that during engine operation only the piston rings touched the upper, or combustion, cylinder sleeve. Two oil seals were installed below the combustion cylinder. The upper seal prevented the oil and gases which leaked past the piston rings from escaping into the crankcase. These products were led out through passages above this seal and thus could be measured. The lower seal helped to prevent the

crankcase oil which lubricated the crosshead from contaminating the lubricant supplied through the junk rings to the combustion cylinder.

Sleeve-motion measuring apparatus. - In order to obtain more detail in the friction records, the diaphragm system was made approximately thirty times as stiff as that used in references 1 and 2 by making the lower diaphragm spring 0.075 inch thick instead of 0.018 inch, as previously used. The upper spring thickness remained at 0.018 inch. The measured natural frequency of this stiff system was about 1100 cycles per second, and the spring rate was approximately 815,000 pounds per inch.

The increase in stiffness made it necessary to use a more sensitive method of measuring sleeve motion. Accordingly, the optical system used previously was removed and an electromagnetic pickup was substituted (fig. 4). This device was connected to an impedance bridge circuit (fig. 5), which was supplied with carrier voltage of 5000 cycles per second. The output from the bridge, which was adjusted both in amplitude and phase to a nearly perfect balance, was amplified and applied to the y-axis deflection amplifier of a DuMont Type 208 oscillograph. During engine operation the motion of the cylinder sleeve changed the position of the armature in the pickup unit and thus changed the balance of the bridge circuit. The edge of the resulting modulated carrier wave was centered on the oscillograph screen and photographed for a permanent record of the sleeve motion. Such photographs were made by turning off the electrical sweep provided in the oscillograph and projecting the image of the trace onto a film which moved with known speed at right angles to the direction of motion of the trace. Figure 6 shows the measuring equipment schematically.

This method of recording the sleeve motion proved to have adequate sensitivity for measuring the extremely small deflections involved (20 or 30 microin.). Calibrations were made by loading the sleeve-diaphragm assembly with test weights and recording the trace deflections observed on the cathode-ray tube. The celluloid cross-section screen which was inserted at the tube face for this purpose was left in place when photographs were made during engine operation. The resulting horizontal lines appearing in the friction records establish a force scale which is convenient in interpreting the results. The top dead-center position of the engine crankshaft was established by a neon lamp which flashed once per revolution of the engine crankshaft. Light from the lamp illuminated a slit, and an image of the slit was focused on the back of the film as it passed through the film gate in the camera.

Other Apparatus

Fuel-air mixture was supplied to the engine from a steam-jacketed vaporizing tank, and the exhaust gases were cooled in a surge tank before passing into the laboratory exhaust system. Inlet and exhaust pressures were measured in these tanks. Temperatures were measured with mercury-in-glass thermometers with the exception of the lower crankcase oil supply, which was measured by a vapor-pressure thermometer. All accessory fuel, water, and oil pumps were driven by electric motors. Engine torques were measured by means of a cradled electric dynamometer and a hydraulic scale.

PROCEDURE

Run-in Tests

The first part of the experiments consisted of run-in tests to determine the variation of ring friction with increasing running time. Three combinations of rings and cylinder-sleeve materials were used in these tests as indicated in the following table. The numbers 1 to 5 in the second column refer to location of the rings on the piston, the top ring being number 1.

Cylinder-sleeve material	Rings
A. SAE 4140	1. Straight-faced compression 2. Straight-faced compression (fig. 7) 3. Straight-faced compression 4. Beveled oil scraper (fig. 8) 5. No ring used
B. SAE 4140	1. Porous chrome-plated straight-faced compression (fig. 9) 2. Straight-faced compression 3. Straight-faced compression (fig. 7) 4. Beveled oil scraper (fig. 8) 5. No ring used
C. Van der Horst porous chrome plate	1. Same as in A 2. Do. 3. Do. 4. Do. 5. Do.

The rings used were of cast iron with a Rockwell hardness on the D scale of 40-50 and the following approximate composition:

Total carbon, percent	3.50-3.80
Silicon, percent	2.20-3.10
Sulphur, percent	0.10 max.
Phosphorous, percent	0.15-0.40
Manganese, percent	0.40-0.80
Molybdenum, percent	0.50-0.70
Chromium, percent	0.20-0.40
Copper, percent	0.50-0.75 (according to section)

The piston rings used are shown in figures 7 to 9. The tension, gap, and surface-finish data for individual rings appear in tables I and II. Diametral ring tensions were measured before and after each run-in test by means of the device described in reference 2.

Before each run the SAE 4140 barrel was lapped 50 strokes with 600 emery using an old piston and cast-iron rings as a tool. After thorough cleaning, a lacquer replica of the surface was made, and the surface condition was measured with a profilometer. The porous chrome barrel was not lapped, but was run in the condition as received from the manufacturer, that is, with a honed surface.

Before the combustion-cylinder sleeve assembly was installed on the engine, the motion measuring apparatus was calibrated by loading with test weights.

Prior to each run-in test the apparatus was thoroughly cleaned and flushed with fresh oil. After assembly the run-in test was made with the following conditions:

Engine speed, rpm	1200
Fuel-air ratio	Best power
Spark advance	Best power
Manifold pressure, in. Hg abs.	28
Crankcase oil temperature, $^{\circ}$F	185 ± 5
Oil temperature at inlet to bearings, $^{\circ}$F	180 ± 2
Crosshead cylinder temperature and combustion cylinder temperature, $^{\circ}$F	180 ± 1
Inlet mixture temperature, $^{\circ}$F	150 ± 3
Oil pressure, lb/sq in.	45
Lubricating oil	SAE aircraft grade, no additives
S.S.U. at 100° F	331
S.S.U. at 210° F	53
Specific gravity at 60° F	0.88
A.P.I. degrees at 60° F	29.3

(Oil for all runs was taken from the same barrel.)

The engine was run for 1 hour under these conditions, and friction records were photographed approximately every 10 minutes. The combustion cylinder was then removed and cleaned. Replicas were made of the cylinder and ring surfaces, and profilometer readings made of the cylinder surface. The motion measuring apparatus was recalibrated.

The run-in test was then continued for nine more hours under the listed conditions with friction records being photographed approximately every hour. At the end of this period the cylinder and rings were removed and measured and the motion measuring apparatus recal ibrated. This procedure was repeated for each of the three combinations of ring and barrel materials used in these tests. Data from these runs appear in table I.

Other Tests

In addition to the run-in tests, a supplementary series of tests was performed to determine the effects of engine speed, load, and cylinder temperature on piston-ring friction.

For these tests the SAE 4140 barrel and ring combination, listed as A in the first table under Procedure, was used. The cylinder was lapped and the other measurements of rings and barrel were made as before. The combination was then run-in as in the previous tests, and data were taken to serve as a check on the original run-in test. However, the friction records photographed during this check run-in were of no value quantitatively, since the calibration of the measuring apparatus was uncertain because of the presence of moisture which leaked into the electromagnetic pickup unit during the test. Nevertheless, the barrel and rings at the end of this test were in a run-in condition, and it was assumed that further changes in friction due to changes in barrel and rings would be minor.

After inspection and measurement of the run-in barrel and rings, the apparatus was calibrated, reassembled, and run for an hour to establish temperature equilibrium. Friction records were than made with the engine running full throttle (manifold absolute pressure, 28 in. Hg abs.) and throttled (20 in. Hg abs.) at a speed of 1200 rpm. Next, full-throttle runs were made with cylinder jacket temperatures of 203°, 180°, 150° and 100° F. Following these runs the engine speed was varied from 1400 to 310 rpm at full throttle and a cylinder jacket temperature of 180° F. The engine was then disassembled and the measuring apparatus calibrated. The results of these tests appear in table II.

PRECISION OF FRICTION MEASUREMENT

A static calibration of the friction measuring apparatus was made before and after each run. If the second measurement was different from the first by more than 2 percent, the run was rejected. The values shown in tables I and II are the averages of measurements made before and after the run in each instance.

A possible source of error was introduced by the junk rings on the cylinder head. If these rings became fouled and stuck during operation, the friction record became greatly altered in appearance and showed large discontinuities and less evidence of sleeve vibration at its natural frequency (because of the damping introduced). Comparatively little difficulty was experienced from this cause, however, and it is believed that the records used in this report are free from these distortions.

The determinations of friction mean effective pressure made from measurements of any one cycle are, on the average, reproducible within ±2 percent. Variations in the shape of the friction record from one cycle to the next on a given film usually do not appear to be significant even when the film is enlarged many times.

RESULTS AND DISCUSSION

Run-in Tests

Figure 10 shows the values of piston-ring mean effective pressure plotted against running time. The piston-ring mean effective pressure was measured from enlarged friction records in the manner described in reference 2, and is defined as

$$\text{fmep} = \frac{\text{Friction work for 1 revolution, in.-lb}}{\text{Piston displacement, cu in.}}$$

The data are taken from table I. It is apparent that no large changes in ring friction occurred at any time during the running-in period. It would appear, however, that there is a rather definite trend toward slightly reduced friction during the first hour or so of running in the case of runs in tables I(a) and I(b). The values of friction work computed from records made during this period for the chrome-plated cylinder (table I(c)) are not considered quantitatively reliable and are not shown.

The data for the remainder of the run-in period show a rather random scattering. The cause of this effect is not known, but it may be due to errors of various kinds (mentioned previously under Precision of Friction Measurement) or to real variations in engine conditions which were not measured in this investigation. The investigation reported in reference 3 mentions a slow rotation of piston rings observed during motoring tests with a glass cylinder, and suggests that the resulting variations of quantity of oil on the cylinder and piston may account for the otherwise unexplained periodic variations in the friction of a running engine. It should be noted, however, that after the first hour the total ring-friction work for each of the runs of figure 10 seldom deviates from constant by more than 0.5 pound per square inch equivalent friction mean effective pressure. It appears, in general, though, that the chrome-plated barrel operated with all cast-iron rings had the greatest friction, and the SAE 4140 barrel operated with all cast-iron rings the lowest. The SAE 4140 barrel with a chrome-plated top ring gave the most erratic results, but the friction work on the average is intermediate between the other combinations.

It is of interest also that the friction work for the compression and expansion strokes is considerably more than that for the exhaust and inlet strokes. This difference is probably due to higher gas pressure behind the piston rings or possibly to differences in oil film temperature, or changes in distribution of oil on the cylinder wall. Some evidence of the effect of gas pressure on lubrication is found in the tests reported in reference 3 which were made with a glass cylinder and show that the gas pressure in the engine cylinder influenced the amount of lubricant on the piston skirt. Examination of the friction records of figures 11 to 21 shows that the friction force is greater during the combustion process, when the gas pressure in the cylinder is much greater than during other parts of the cycle.

The friction records for the tests are shown in figures 11 to 21. For these records the film speed was 25 inches per second. The vertical lines extending across the photograph are index lines showing the top dead-center position of the piston. With the exception of record 1E the first stroke appearing at the left is the suction stroke, followed by the compression, power, and exhaust strokes. More than 2 revolutions are usually represented. The closely spaced horizontal lines can be used as a force scale, the value of which is shown in each instance in tables I and II under Remarks.

The values of friction mean effective pressure given in table I and in figure 10 are considerably less than those reported in reference 2. The reason for this difference is probably that the values in reference 2 include piston-skirt friction; while the crosshead used in the present investigation prevented any contact between the piston and the cylinder wall. Also, the apparatus of reference 2 provided normal lubrication of the cylinder wall; while the engine with the crosshead received all cylinder lubrication from the cylinder-head end through the junk rings. The effect of this difference in lubrication is not known because of confusion with other effects, such as that mentioned previously. The piston rings used in the present investigation had tensions intermediate between those of the two types of ring used in the work reported in reference 2.

The tests of reference 2 showed a more pronounced decrease in friction during the running-in period than the tests of the present investigation. It is possible that, in the engine without a crosshead, the cocking of the piston during running caused the ring surfaces gradually to become slightly convex. This convexity should favor the formation of an oil film between piston ring and cylinder wall with a resulting decrease of friction.

A peculiar effect was produced when the engine was motored. (See the points marked M in fig. 10.) The ring friction work during the compression and expansion part of the cycle was reduced, but the friction during the exhaust and inlet strokes increased slightly. The reduction of friction during the expansion part of the cycle can be explained on the basis of reduced gas pressures behind the rings when combustion is absent. A careful examination of enlarged friction records showed that the increase in motoring ring friction during the low pressure strokes of the cycle could not be attributed definitely to either stroke alone; apparently there are small increases in friction during both strokes. These increases may be due to lowered oil film temperatures or to changes in supply or distribution of lubricant.

Examination of the friction records shows that the natural frequency of the measuring system is high enough to give a good indication of sleeve motion even at the ends of the stroke, where an abrupt change in force is evident. The fact that there is a sudden change of considerable magnitude is good evidence that there is some coulomb friction when the piston velocity is low. The height of these vertical discontinuities is a measure of the breakaway value of the static friction between ring and cylinder wall under the various conditions of engine operation. The presence of this type of friction is probably, at least in part, the source of the excitation of the sleeve vibrations at natural frequency which appear in all the records. Examination of records 1G through 4G (fig. 22) will show, however, that the apparatus registered some sleeve excitation even when motored with no piston rings (cylinder head removed). The cause of this excitation is undoubtedly the various mechanical impacts and general vibration of the engine.

The records of figure 19 indicate that the coulomb friction force at the ends of strokes was less pronounced when the engine was throttled (2D) than when operated with full throttle (1D). Record 1F (fig. 21) shows that this sudden change in force is smaller when the jacket temperature is 100° F than when the jacket is operated at higher temperatures (records 2F, 3A, and 4F), although the friction work is increased at the lower temperature (table II and fig. 23).

Records 1E and 4E of figure 20 indicate that the breakaway friction is highest for the lowest and highest speeds used; while the intermediate engine speeds show lower values of breakaway friction (records 2E and 3E). At the very lowest speed the partial oil film at the end of the stroke is probably thinner than at higher speeds. At the highest speed the changes in supply and distribution of oil plus the possibility of higher local film temperatures may combine to decrease the film thickness, with a consequent increase in breakaway friction.

Effect of Load

Figure 19 shows the friction records taken with the engine operating at two different inlet pressures and figure 23 shows the corresponding friction mean effective pressure. The reduction in friction with lower inlet pressure is due largely to smaller friction forces during the combustion stroke. This effect is probably due to a reduction in gas pressure behind the piston rings. The investigation reported in reference 4 yielded a similar result when ring loading was varied.

Effect of Engine Speed

The effect on piston-ring friction of varying engine speed is shown in figures 20 and 23. Here the friction work changes in the following manner:

rpm	fmep, exhaust-inlet	fmep, compression-expansion	Variation of rpm	Proportional increases		
				rpm	fmep, exhaust-inlet	fmep, compression-expansion
800	1.23	3.07	800-1200	1.5	1.4	1.16
1200	1.74	3.59	1200-1400	1.16	1.6	1.25
1400	2.74	4.48	800-1400	1.75	2.2	1.45

If the rings operate with fluid film lubrication and constant film thickness and oil viscosity, simple hydrodynamic theory would require that the friction work increase in proportion to the speed. If the rings operate with no lubrication, the friction work should not change with speed. The changes in work for the compression and combustion strokes appear to fall between these limits. There is indication, however, that the friction work during the exhaust and inlet strokes increases too rapidly with speed for the constant-fluid-film theory. It can be supposed, therefore, that with increasing speed the film thickness during these strokes is decreased. This decrease is probably due to insufficient supply or distribution of lubricant on the cylinder wall, although there is also the possibility of increased local film temperatures at higher speeds. This would decrease viscosity and tend to decrease film thickness.

Effect of Jacket Temperature

The effect of changing the cylinder coolant temperature is shown in figures 21 and 23. An increase in temperature from 100° to 200° F reduced the total ring friction mean effective pressure from 7.64 to 5.35. The work for the inlet and exhaust strokes decreased slightly more than that of the high-pressure strokes of the cycle. This effect can be attributed to decreased viscosity of the oil on the cylinder surface, and suggests that there may be more nearly fluid film lubrication when the gas pressure behind the rings is small.

In evaluating the effects of changes in load, speed, and temperature, it is well to remember that even after the engine had been run continuously for several hours under constant conditions, there were random variations in friction which are not explained (fig. 10). This fact may have tended to distort the actual values of friction measured while a study of other variables was being made. However, the trends appear to be reasonable.

Other Results

Blow-by.- The measurements of blow-by rate under various conditions of operation show a random variation. (See tables I and II.) It is not surprising, however, that the lowest rate recorded during the entire investigation occurred during operation of the engine with low manifold pressure. There is some evidence that blow-by is less at reduced engine speeds (table II). This observation is not a certainty, however, because of the variations observed during operation under "constant" conditions.

<u>Oil consumption</u>.– The amount of oil which passed by the piston rings and was caught by the upper crosshead oil seal was usually a very small quantity – too small, in fact to permit a very accurate measurement because of the relatively large amount which clung to the engine parts. In the case of the porous chrome-plated barrel, however, the quantity was large. It is estimated that the amount was probably 10 times the quantity which leaked past the rings during operation with the other ring-barrel combinations tested.

Figures 24 to 31 are photomicrographs made from replicas of the ring and barrel materials used in the tests. These show the condition of the surfaces at the beginning, after 1 hour, and after 10 hours of running. The ragged edges of the ring photographs do not represent the edge conditions of the ring, but are due to slight tearing of the lacquer film when it was lifted from the surface.

CONCLUSIONS

Under the conditions of the tests reported in this investigation, the results obtained point to the following conclusions:

1. Piston-ring friction decreased slightly with increasing running time, the greater part of the change occurring during the first hour.

2. The cast-iron piston rings operating in a SAE 4140 barrel had the lowest friction of the combinations tested. The cast-iron rings in a porous chrome barrel had the greatest friction, and the SAE 4140 barrel with one chrome top ring had intermediate friction. These differences, however, were small.

3. Piston-ring friction increased with engine speed.

4. Piston-ring friction decreased with increased cylinder jacket temperatures.

5. Lowering the manifold pressure reduced piston-ring friction.

6. The oil flow past the piston rings, from the cylinder head toward the crankcase, was much larger when the engine was operated with a porous chrome-plated barrel and all cast-iron rings than during operation with the other ring-barrel combinations tried.

Sloan Laboratories for Aircraft and Automotive Engines,
Massachusetts Institute of Technology,
Cambridge, Mass., June 18, 1946.

REFERENCES

1. Forbes, J. E., and Taylor, E. S.: A Method for Studying Piston Friction. NACA Wartime Rep. W-37 (Originally issued as NACA ARR, March 1943).

2. Leary, W. A., and Jovellanos, J. W.: A Study of Piston and Ring Friction. NACA Wartime Rep. W-47 (Originally issued as NACA ARR No. 4J06, 1944).

3. Shaw, Milton C., and Nussdorfer, Theodore: Visual Studies of Cylinder Lubrication, Part I. The Lubrication of the Piston Skirt. NACA Wartime Rep. E-66 (Originally issued as NACA ARR No. E5H08, 1945).

4. Tischbein, Hans W.: The Friction of Piston Rings. NACA TM No. 1069, 1945.

TABLE I.– RUN-IN TEST

(a) Cylinder, SAE 4140; all rings, cast iron

Running time (min)	Film	Brake load (lb)	Blow-by (cu ft/hr) average since previous reading[1]	Piston ring fmep (lb/sq in.) Compression-combustion	Exhaust-inlet	Total	Remarks
0			Indeterminate				Start
7	1A	16.6	----	3.63	2.37	6.00	Scale of friction records, 2.06 lb/line
15	2A	17.7	----	3.73	2.33	6.06	
24	3A	17.5	----	3.59	2.08	5.67	
33	4A	17.4	----	3.54	2.11	5.65	
44	5A	17.7	----	3.62	2.26	5.88	
58	6A	----	2.68	3.63	2.13	5.76	End of 1-hour run
58			Indeterminate				Start of 9-hour run
85	7A	17.9	----	3.31	2.07	5.38	
148	8A	18.6	2.23	3.44	2.04	5.48	
205	9A	18.0	1.95	3.12	2.03	5.15	Scale of friction records, 2.00 lb/line
270	10A	18.0	1.90	3.28	1.60	4.88	
325	11A	18.55	1.94	3.26	1.60	4.86	
387	12A	18.6	2.27	3.66	2.03	5.69	
444	13A	18.6	2.25	3.43	1.69	5.12	
504	(2)	18.5	1.85	(2)	(2)	(2)	
564	(2)	17.7	1.99	(2)	(2)	(2)	
595	(2)	18.7	1.89	(2)	(2)	(2)	End of 9-hour run

[1]Average blow-by for 9-hour run, 2.07 cu ft/hr.
[2]Friction records spoiled in processing.

Before 1-hour run:

Cylinder profile – longitudinal 14 microinches

Piston rings	Diametral tension (lb)	Gap	Profile (microinches)
1-compression	4.69	0.012	22
2-compression	4.82	.012	20
3-compression	4.69	.013	23
4-scraper	9.13	.013	27

After 1-hour run = before 9-hour run:

Cylinder profile – longitudinal 11 microinches

Replicas made of rings, but no other measurements made, since rings were not removed from piston.

Oil.– 8 grams burned or vaporized. A few drops passed the piston rings and were caught by upper oil seal.

After 9-hour run:

Cylinder profile – longitudinal 5.5 microinches

Piston rings	Diametral tension (lb)	Gap	Profile (microinches)
1	4.53	0.012	10
2	4.68	.012	8
3	4.57	.013	10
4	9.42	.013	Not measured

Oil.– 72 grams burned or vaporized. Approximately 10 grams passed the piston rings.

TABLE I.– Continued

(b) Cylinder. SAE 4140; ring 1, chrome plated

Running time (min)	Film	Brake load (lb)	Blow-by (cu ft/hr) average since previous reading[1]	Piston ring fmep (lb/sq in.)			Remarks
				Compression-combustion	Exhaust-inlet	Total	
0			Indeterminate				Start
8	1B	17.3	------	4.28	2.16	6.34	Scale of friction records, 2.07 lb/line
18	2B	17.8	------	4.11	1.87	5.98	
28	3B	18.0	------	3.97	1.89	5.86	
39	4B	18.0	------	3.89	1.77	5.66	
50	5B	18.0	------	3.59	1.85	5.44	
58	6B	18.1	2.23	3.68	1.85	5.53	End of 1-hour run
58		17.6	Indeterminate				Start of 9-hour run
71	7B	17.8	1.88	3.43	2.01	5.44	
121	8B	17.7	1.97	3.47	1.90	5.37	
181	9B	18.0	2.15	3.80	1.93	5.73	Scale of friction records, 2.07 lb/line
242	10B	18.3	2.04	3.51	1.62	5.13	
297	11B	18.0	2.20	3.80	2.13	5.93	
360	12B	18.2	2.19	(2)	(2)	(2)	
421	13B	18.0	2.24	4.13	2.39	6.52	
480	14B	18.2	2.20	3.30	1.91	5.21	
540	15B	18.4	2.37	3.35	1.96	5.31	
596	16B	18.9	2.05	3.15	1.42	4.57	
606	17B	6.0	------	2.20	1.83	4.03	End of 9-hour run

[1]Average blow-by for 9-hour run, 2.15 cu ft/hr.
[2]Film broke – record lost.

<u>Before 1-hour run:</u>

Cylinder profile – longitudinal 13 microinches

Piston rings	Diametral tension (lb)	Gap	Profile (microinches)
1-compression	4.49	0.021	60-100
2-compression	4.79	.012	27
3-compression	4.72	.015	35
4-scraper	9.65	.015	Not measured

<u>After 1-hour run = before 9-hour run:</u>

Cylinder profile – longitudinal 9.0 microinches

Replicas were made of ring surfaces, but no other measurements were made, since rings were not removed from piston.

<u>Oil</u>.– 11 grams burned or vaporized in 1-hour run. A few drops passed the piston rings and were caught by the upper oil seal.

<u>After 9-hour run:</u>

Cylinder profile – longitudinal

Central portion 7 microinches
Top portion 4-5 microinches

Piston rings	Diametral tension (lb)	Gap	Profile (microinches)
1	4.36	0.021	60
2	4.97	.012	12
3	4.57	.015	10
4	10.30	.015	Not measured

<u>Oil</u>.– Approximately 50 grams burned or vaporized in 9-hour run.
Approximately 10 grams passed the rings.

TABLE I.– Concluded

(c) Cylinder, porous chrome plated honed at factory; all rings, cast iron

Running time (min)	Film	Brake load (lb)	Blow-by (cu ft/hr) average since previous reading	Piston ring fmep (lb/sq in.) Compression-combustion	Exhaust-inlet	Total	Remarks
0			Indeterminate				Start of 1-hour run
9	1C	16.6	2.88	(1)	(1)	(1)	Scale of friction records uncertain
20	2C	17.4	1.25	(1)	(1)	(1)	
30	3C	17.3	1.78	(1)	(1)	(1)	
39	4C	17.3	1.71	(1)	(1)	(1)	
49	5C	17.1	------	(1)	(1)	(1)	
58	6C	17.6	1.68	(1)	(1)	(1)	End of 1-hour run
60		15.9	Meter not	(1)	(1)	(1)	Start of 9-hour run
73	7C	16.5	working	3.75	2.16	5.91	
135	8C	17.1	Indeterminate	3.76	2.28	6.04	
191	9C	17.0	1.86	3.91	2.32	6.23	
250	10C	17.2	1.89	4.03	2.34	6.37	Scale of friction records, 1.94 lb/line
310	11C	17.2	1.83	4.06	2.66	6.72	
371	12C	17.4	2.10	4.06	2.28	6.34	
431	13C	17.3	1.88	3.84	2.63	6.47	
489	14C	17.2	1.81	3.63	2.69	6.32	
551	15C	17.4	2.10	3.60	2.80	6.40	
598	16C	17.3	1.98	3.69	2.55	6.24	
602	17C	------	------	3.08	2.80	5.88	Motoring
603							End of 9-hour run

[1]Calibration in doubt. Qualitatively all right.

Before 1-hour run:

Cylinder profile – longitudinal 95 microinches

Piston rings	Diametral tension (lb)	Gap	Profile (microinches)
1-compression	4.42	0.013	35
2-compression	5.02	.016	40
3-compression	4.87	.013	25
4-scraper	9.63	.016	Not measured

After 1-hour run = before 9-hour run:

Cylinder profile – longitudinal 55-90 microinches

Replicas were made of rings but no other measurements were made since rings were not removed from piston.

Oil.– 31 grams burned or vaporized. The amount which passed the piston rings to be caught by the upper oil seal was not measured.

After 9-hour run:

Cylinder profile – longitudinal 40-90 microinches

Piston rings	Diametral tension (lb)	Gap	Profile (microinches)
1	4.34	0.013	13
2	4.89	.016	18
3	4.75	.013	15
4	9.85	.016	15

Oil.– 66 grams burned or vaporized.
122 grams passed the rings and were caught by the upper oil seal.

TABLE II.– FRICTION RECORDS

[Cylinder, SAE 4140, all Rings, Cast Iron]

Film	Engine speed (rpm)	Brake load (lb)	Cylinder jacket temperature (°F)	Manifold pressure (in. Hg absolute)	Blow-by (cu ft/hr)	Piston ring fmep[1] (lb/sq in.)		
						Exhaust-inlet	Compression-expansion	Total
1D	1200	17.4	180	28	2.19	2.37	4.38	6.75
2D	1200	8.5	180	20	1.08	2.06	3.43	5.49
1E	310	19.7	180	28	------	------	------	----
2E	800	17.0	180	28	1.49	1.23	3.07	4.30
3E	1200	18.0	180	28	2.06	1.74	3.59	5.33
4E	1400	17.1	180	28	2.43	2.74	4.48	7.22
1F	1200	14.5	100	28	------	3.04	4.60	7.64
2F	1200	16.8	150	28	2.16	2.28	3.86	6.14
3F	1200	17.7	180	28	2.49	2.37	3.87	6.24
4F	1200	17.5	203	28	2.09	1.84	3.51	5.35

[1]Scale of friction records, 2.00 lb/line.

Rings at beginning of above runs:

Piston rings	Diametral tension (lb)	Gap	Profile (microinches)
1–compression	4.67	0.017	45
2–compression	5.13	.013	50
3–compression	4.68	.015	35
4–scraper	9.42	.011	

Rings at end of above runs:

Piston rings	Diametral tension (lb)	Gap	Profile (microinches)
1–compression	4.32	0.017	8
2–compression	5.00	.013	10
3–compression	4.48	.015	12
4–scraper	9.52	.011	14

Barrel profile after above runs: $8 - 10$ in. $\times 10^{-8}$

WC-5

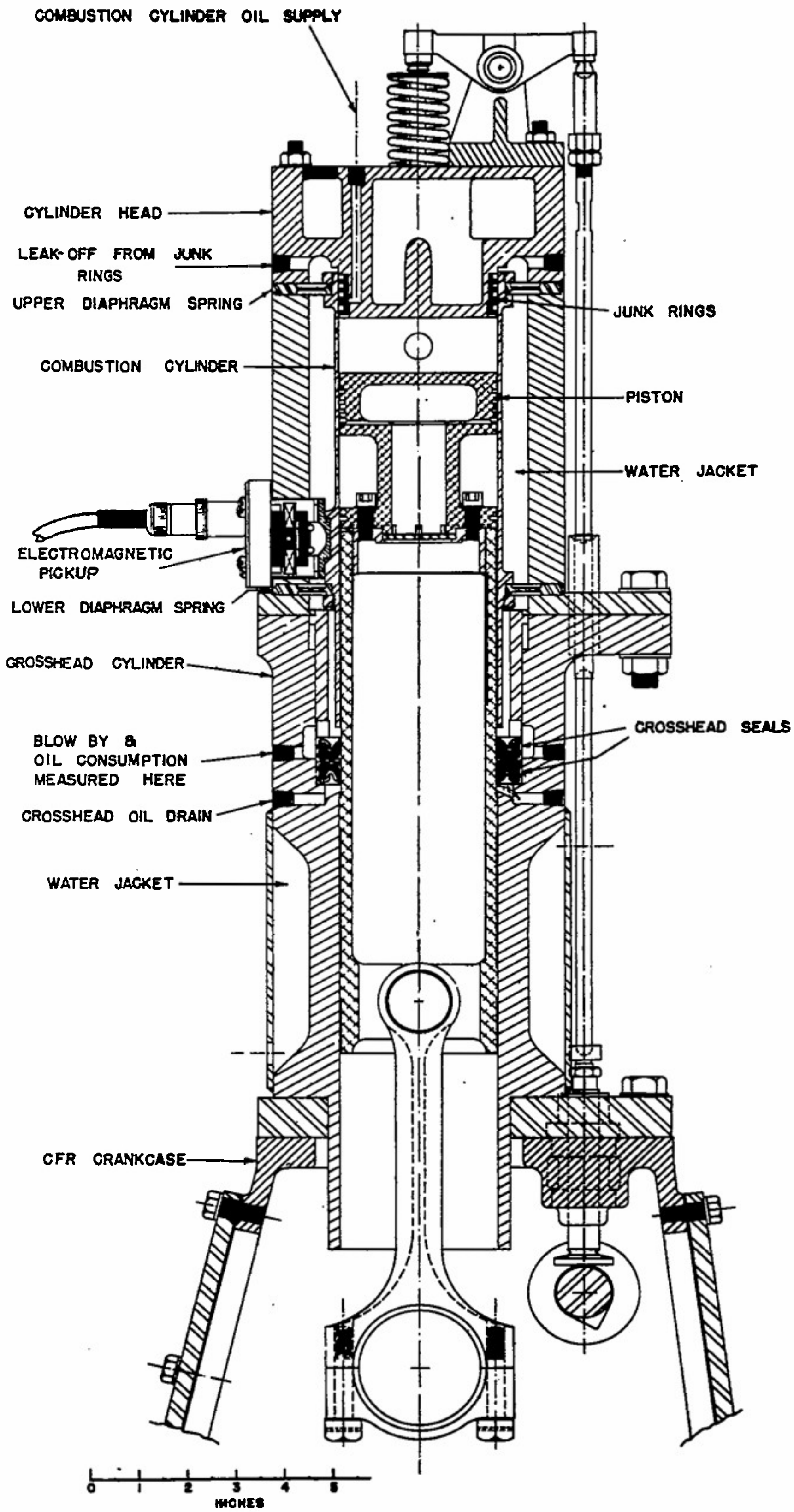


Figure 1.- Engine, showing crosshead and spring-mounted combustion cylinder.

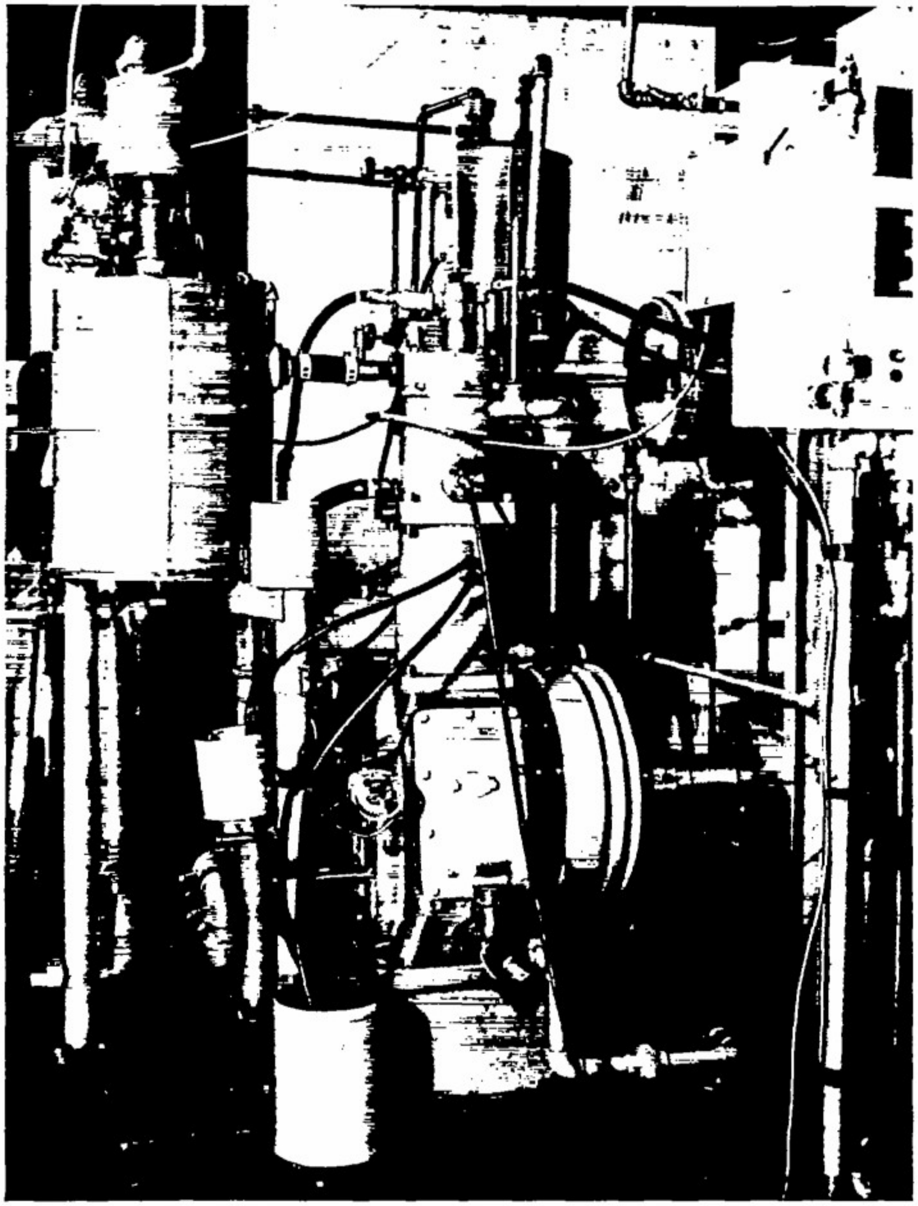
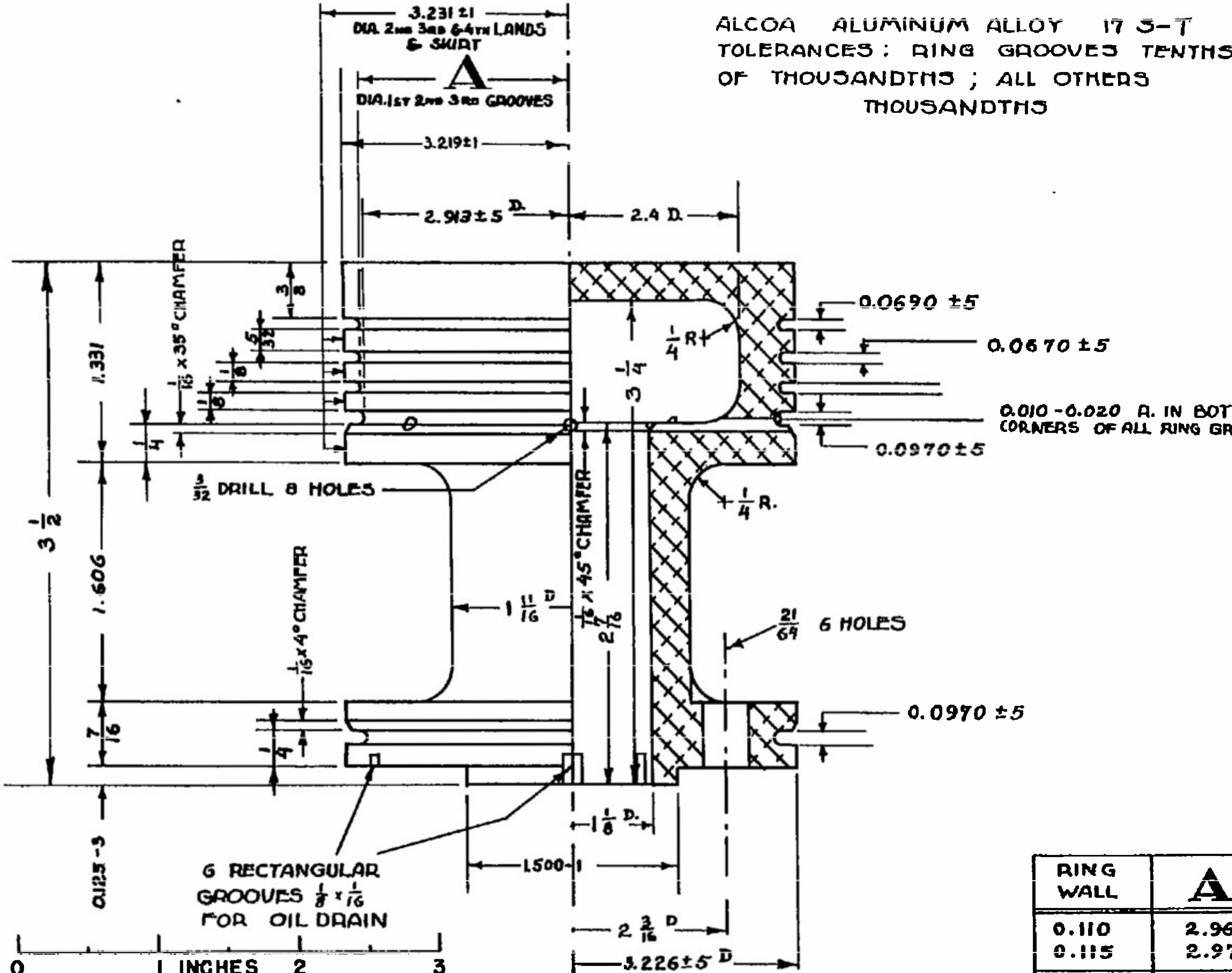


Figure 2.—General view of engine.



RING WALL	A DIA.
0.110 0.115	2.968 2.978
0.157 0.162	2.774 2.784

Fig. 3

Figure 3.- Special piston used in the tests.

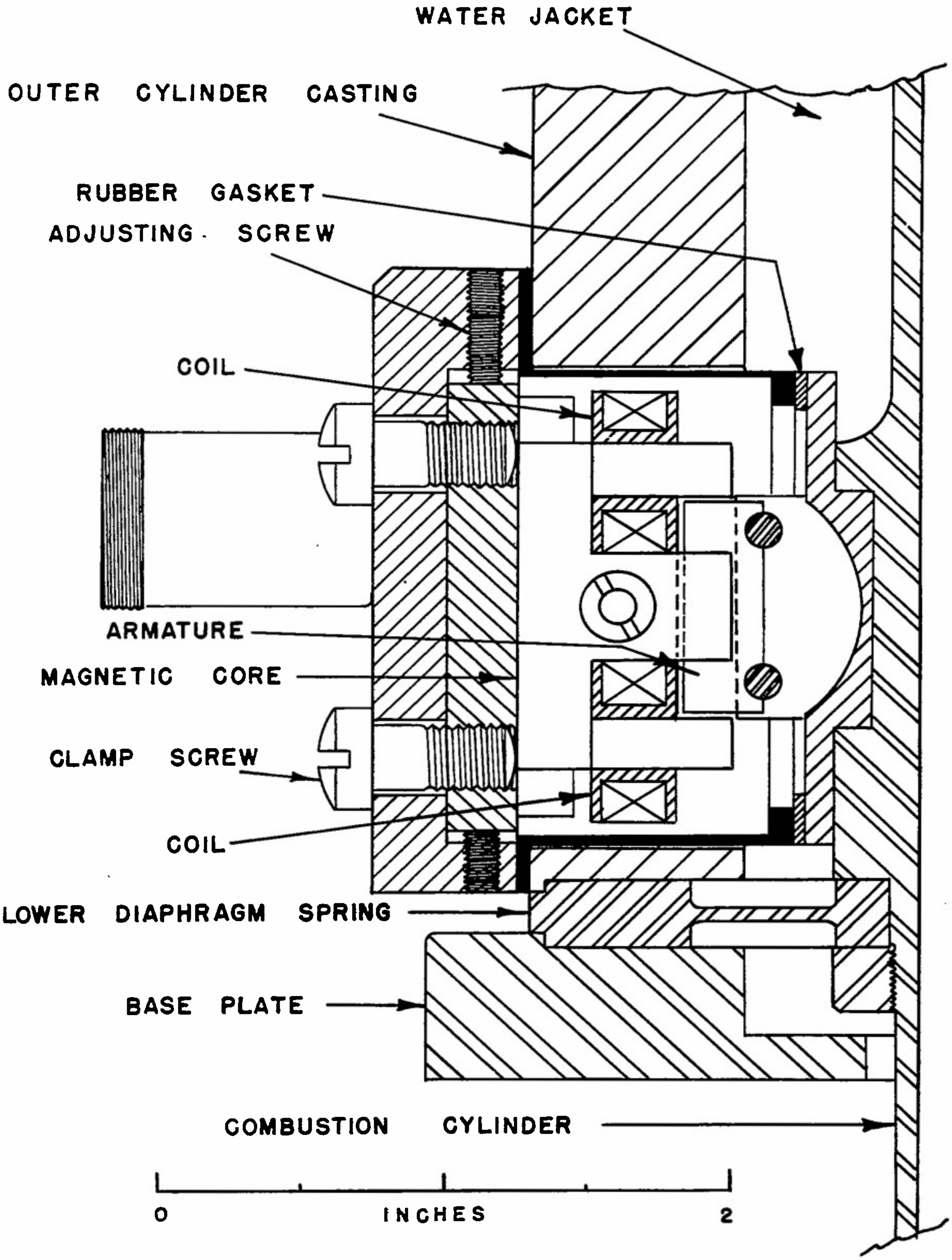


Figure 4.- Details of electromagnetic pickup for measuring cylinder sleeve motion.

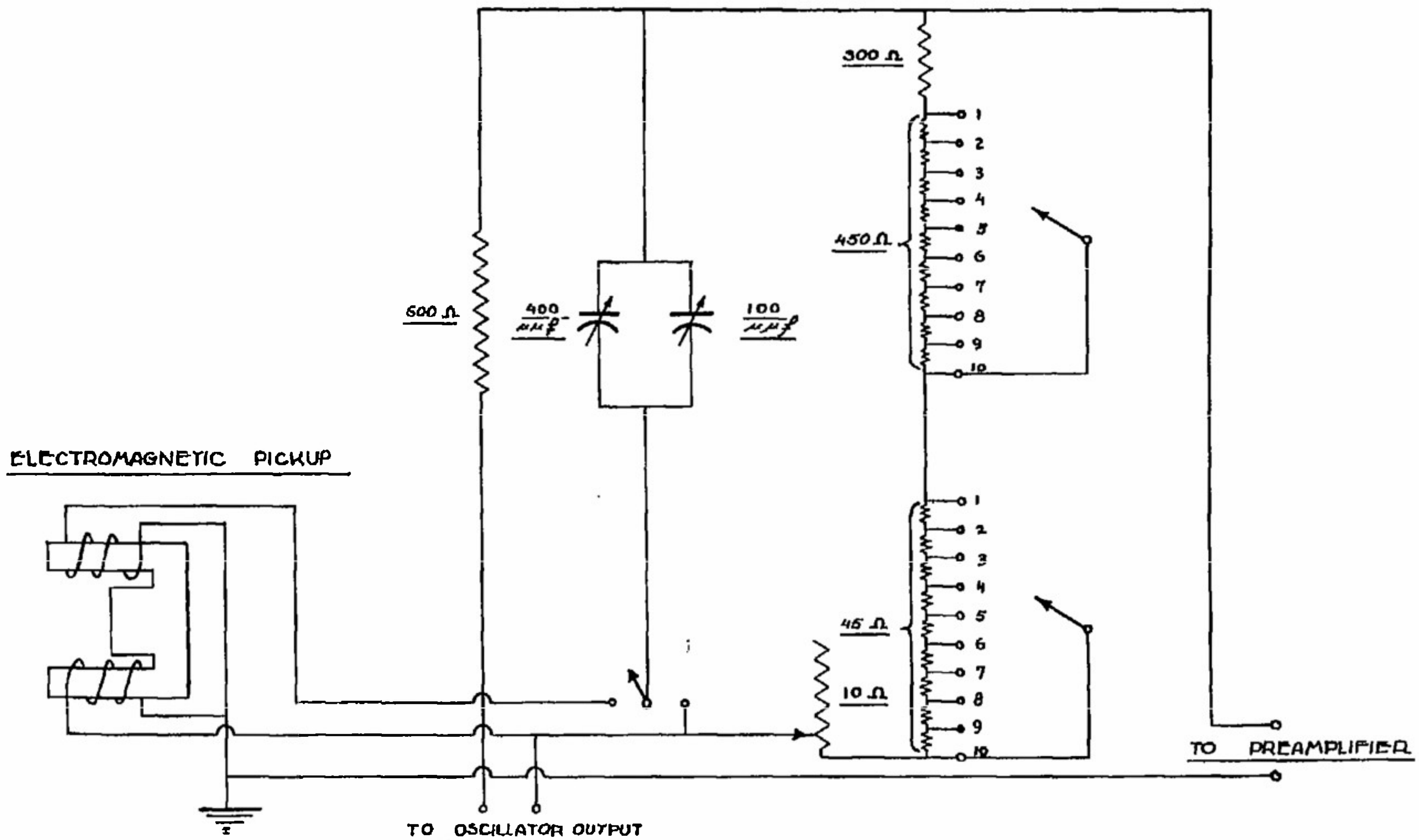


Figure 5.- Electrical diagram of bridge circuit.

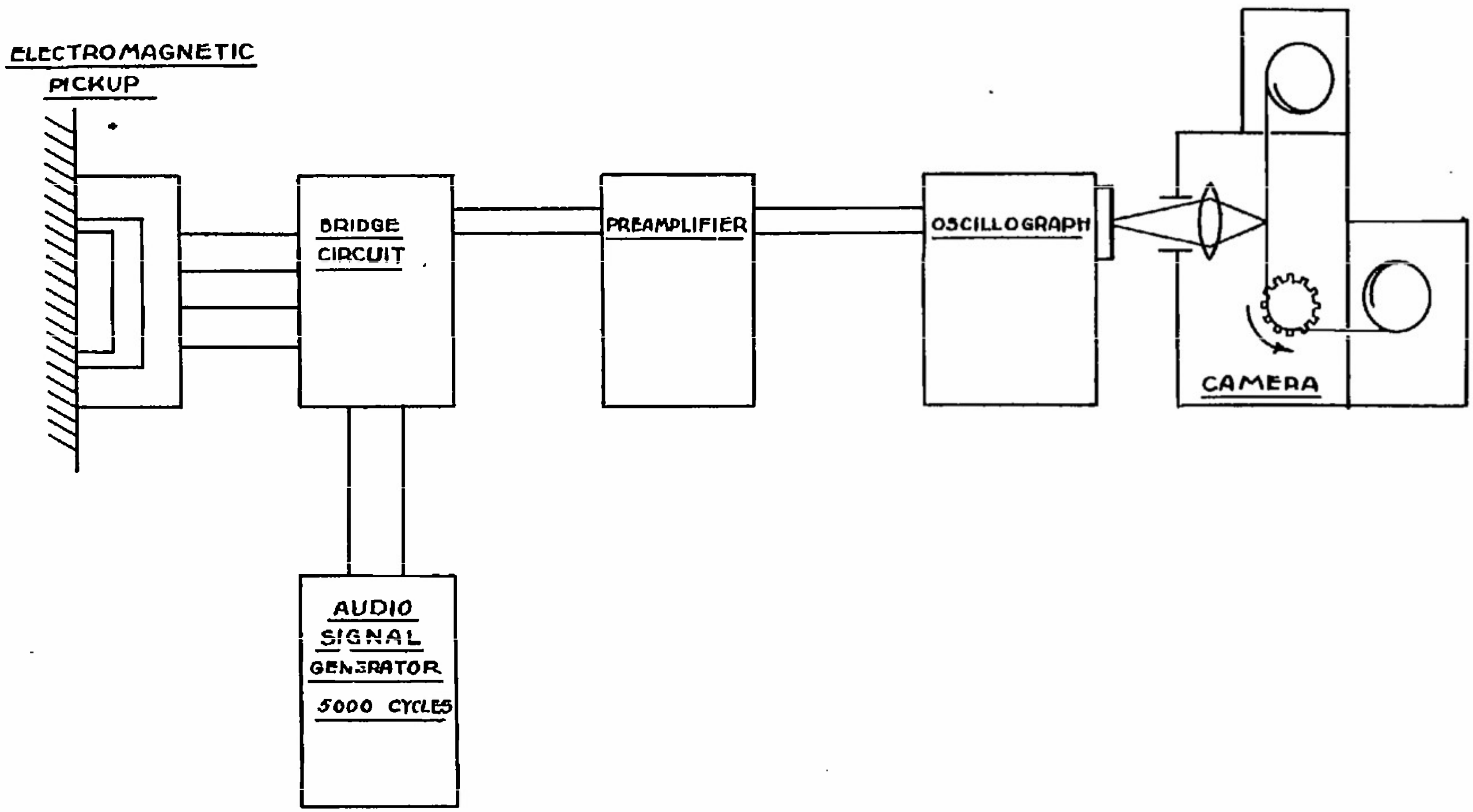


Figure 6.- Schematic diagram of sleeve-motion measuring apparatus.

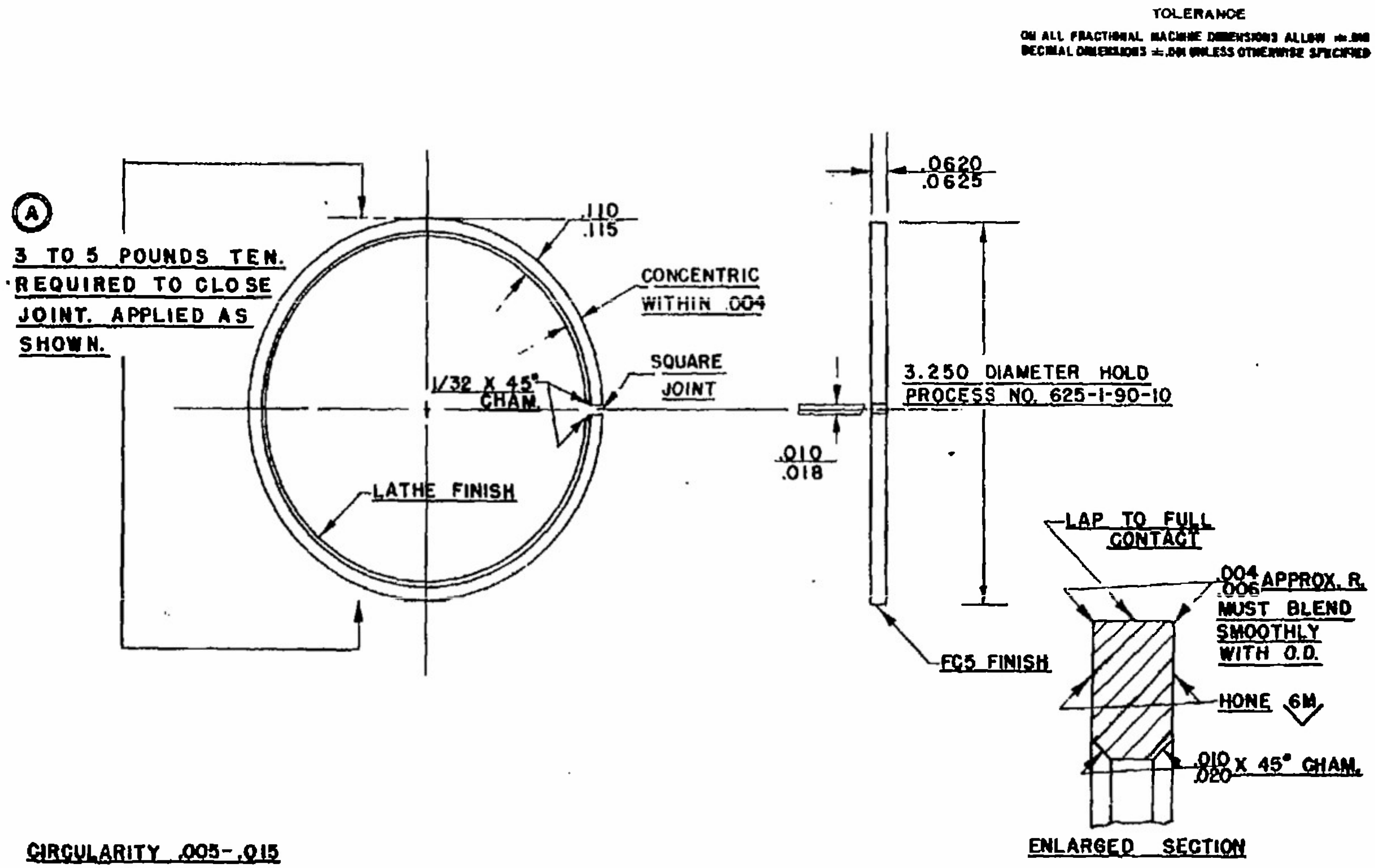


Figure 7.- Plain compression ring used in the tests.

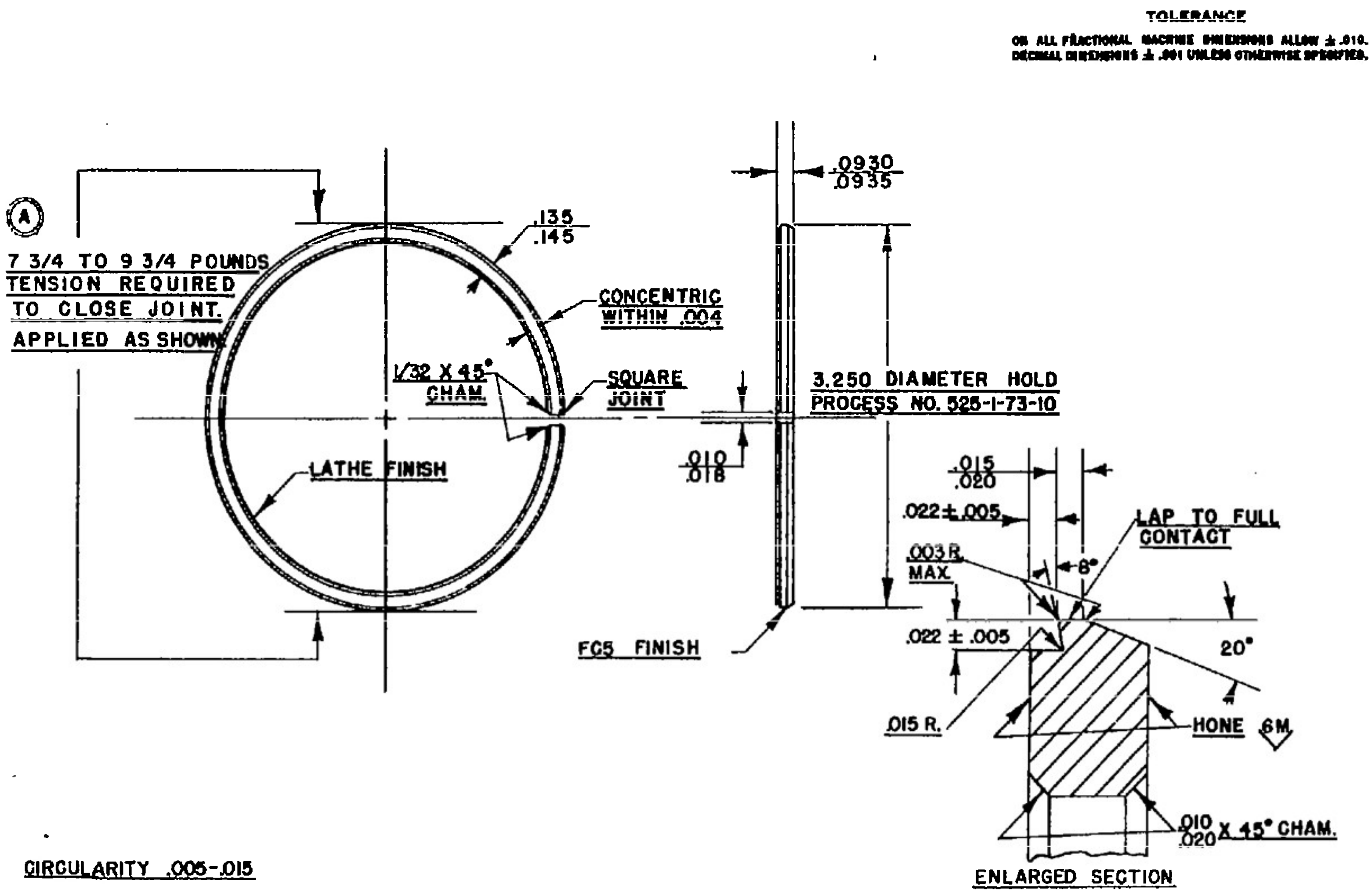


Figure 8.- Beveled oil scraper ring used in the tests.

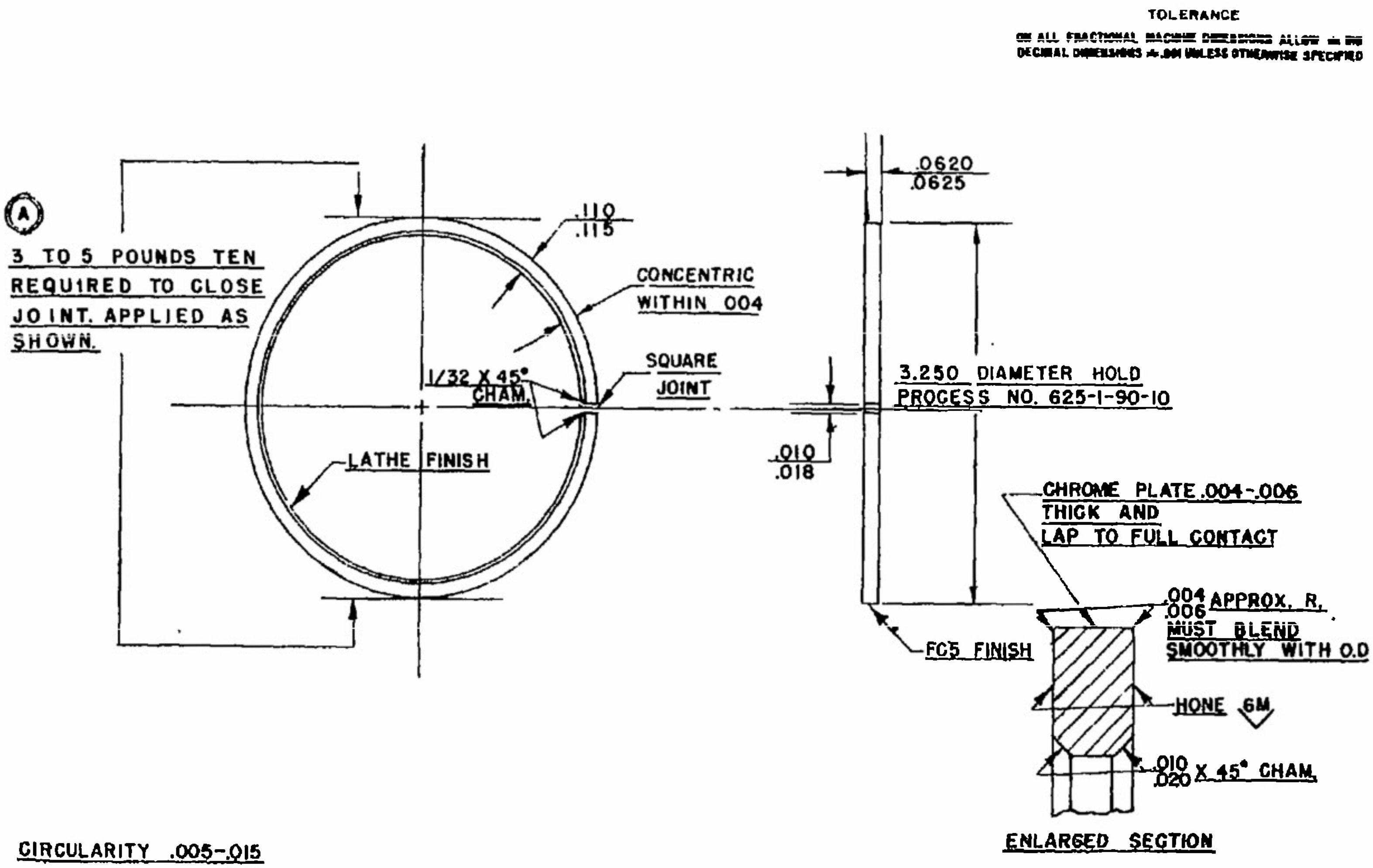


Figure 9.- Porous chrome-plated compression ring used in tests.

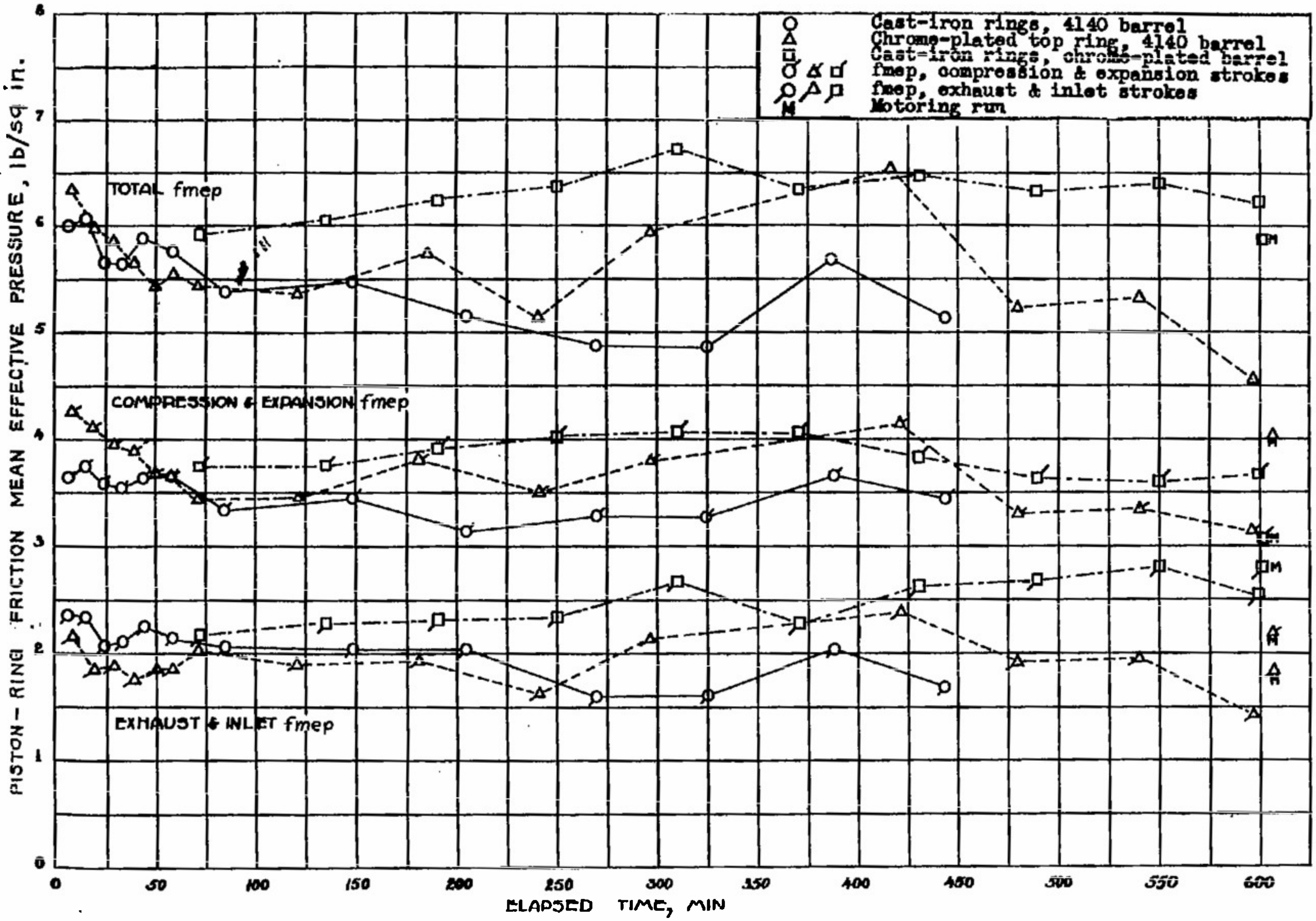


Figure 10.- Effect of running time on piston-ring friction mean effective pressure.

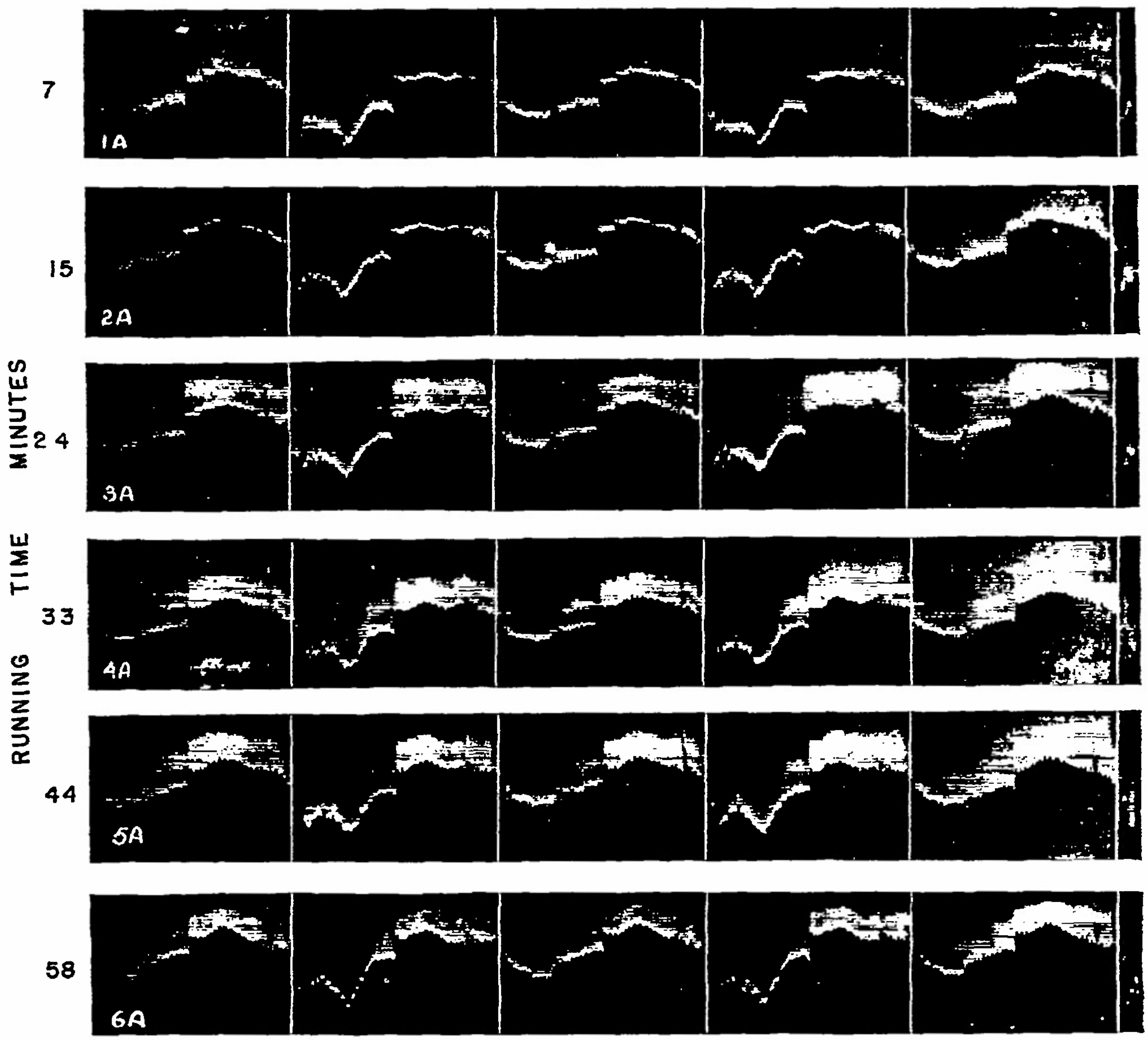


Figure 11.—Friction records for the run-in test with SAE 4140 sleeve and cast-iron rings.

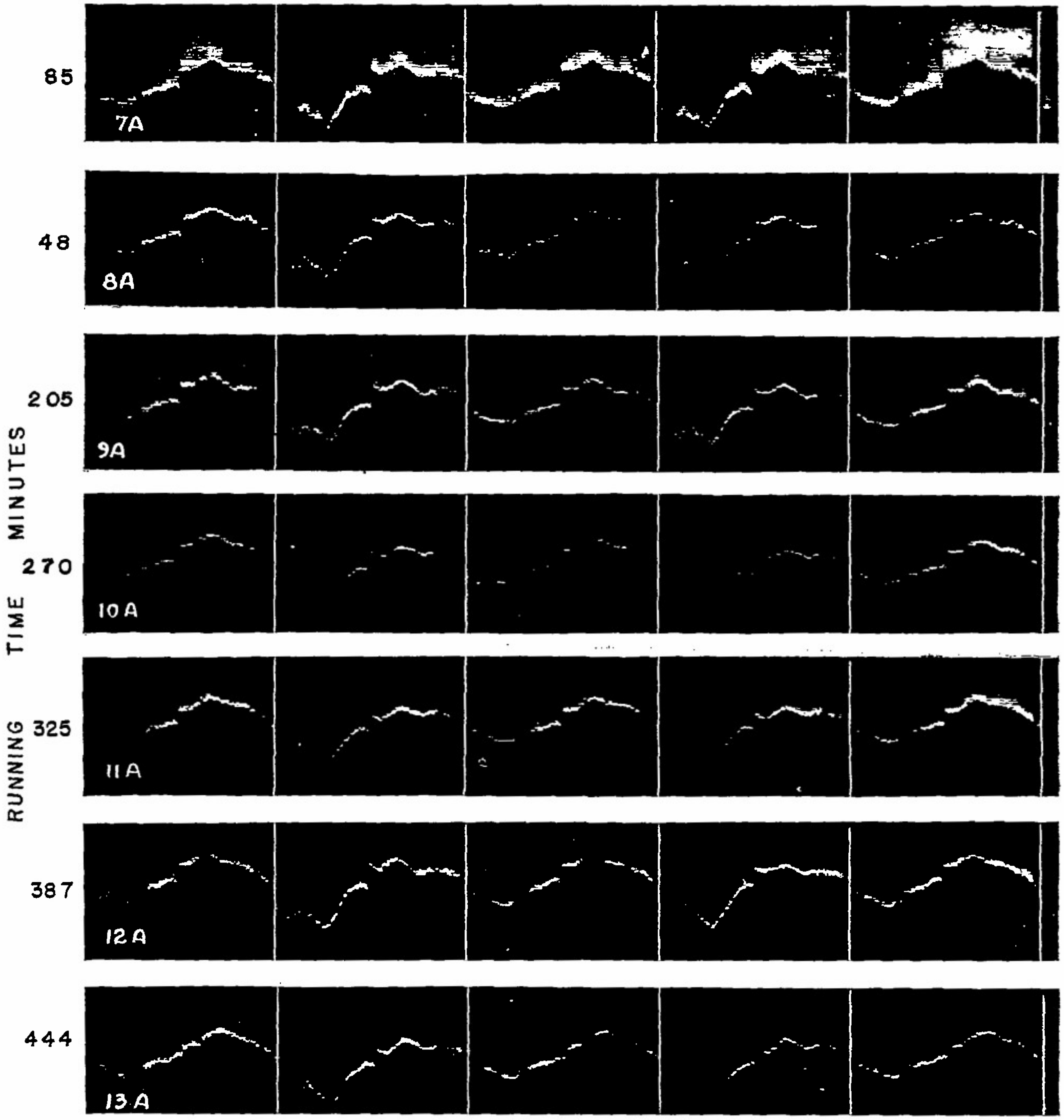


Figure 12.—Friction records for the run-in test with SAE 4140 sleeve and cast-iron rings.

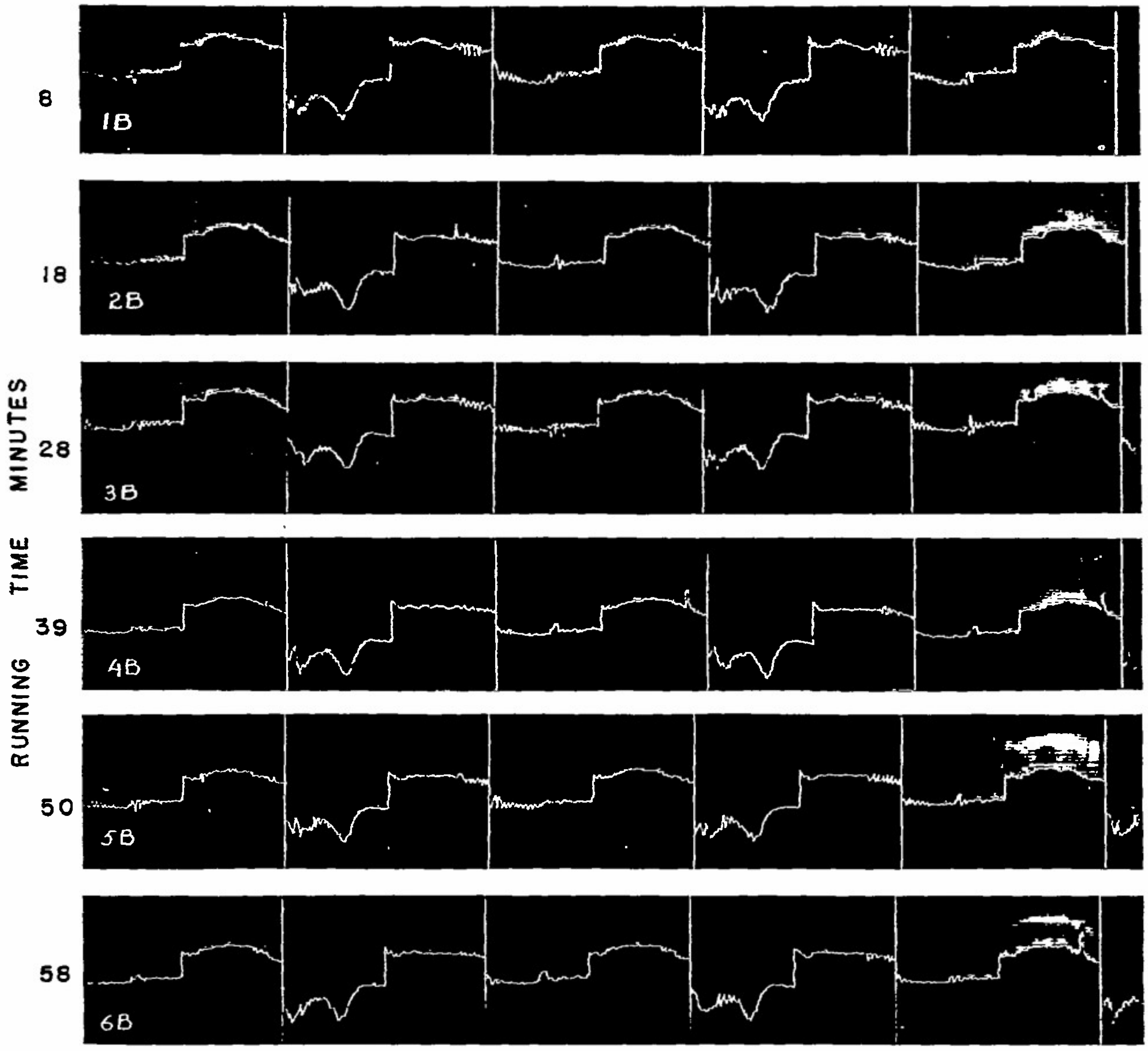


FIGURE 13.—Friction records for the run-in test with SAE 4140 barrel and chrome-plated top ring. (Records have been outlined for clearness.)

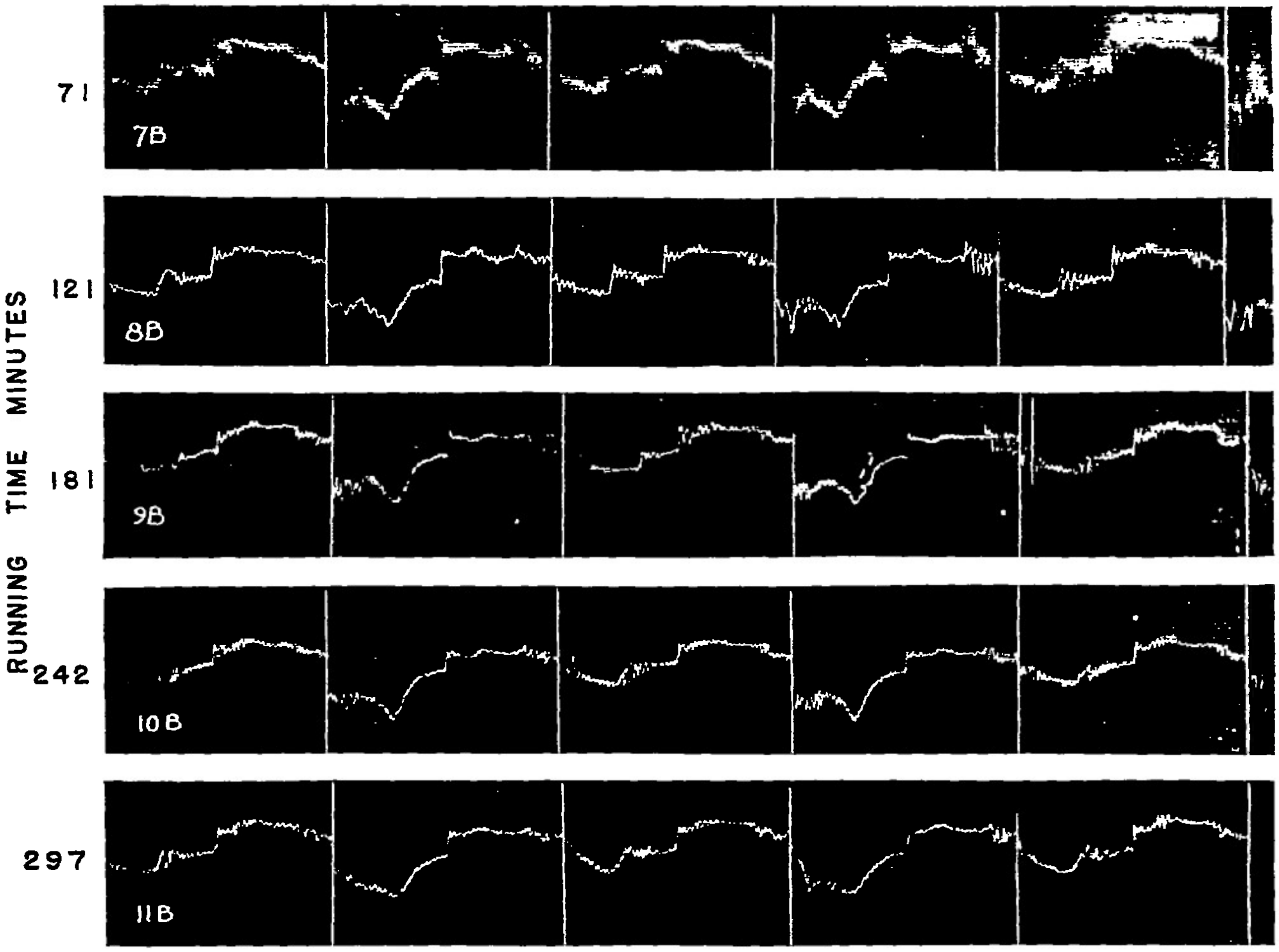


FIGURE 14.—Friction records for the run-in test with SAE 4140 sleeve and chrome-plated top ring. (Records 8B to 11B have been outlined for clearness.)

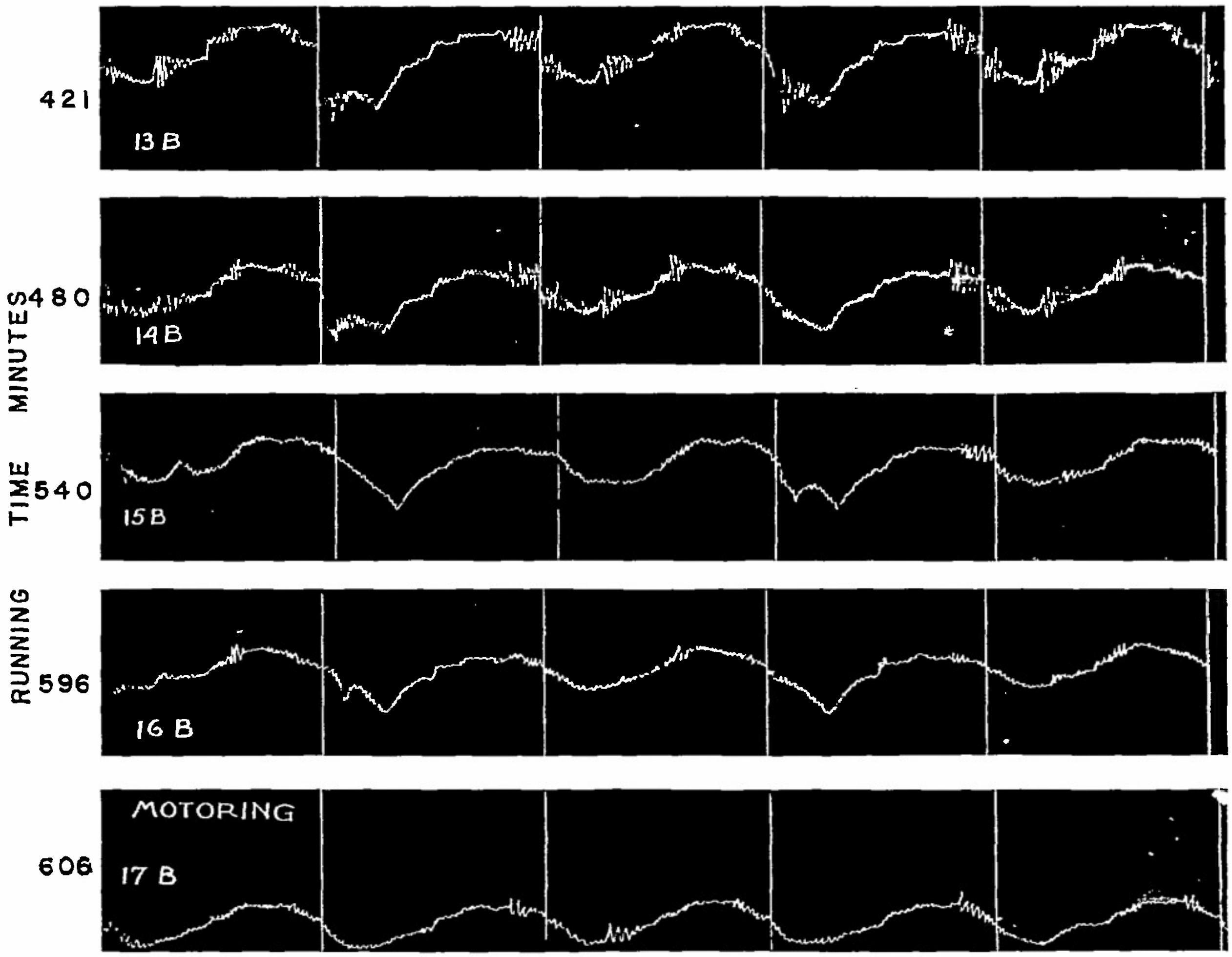


FIGURE 15.—Friction records for the run-in test with SAE 4140 sleeve and chrome-plated top ring. (Records have been outlined for clearness.)

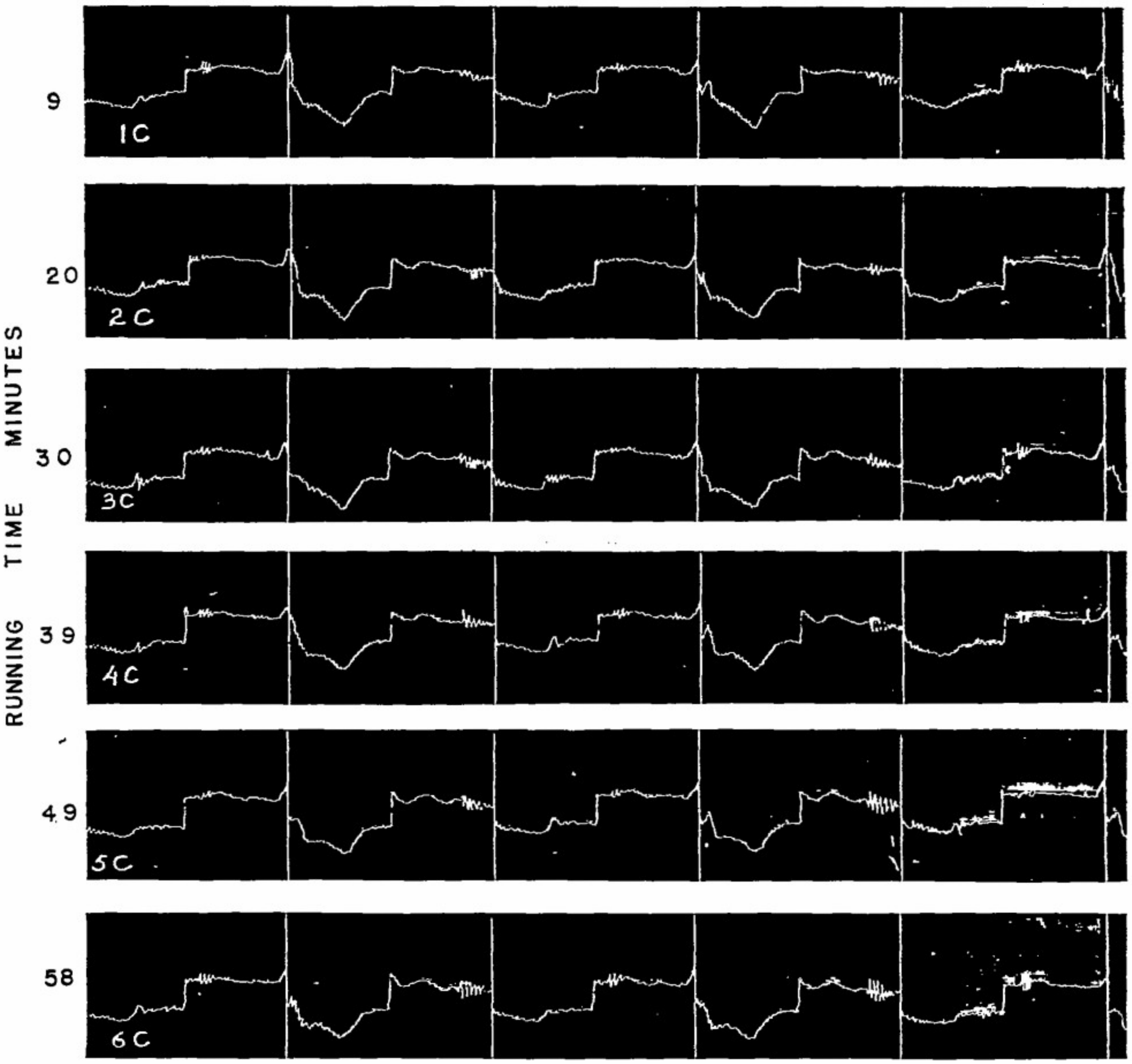


FIGURE 16.—Friction records for the run-in test with porous chrome-plated sleeve and cast-iron rings. (Records have been outlined for clearness.)

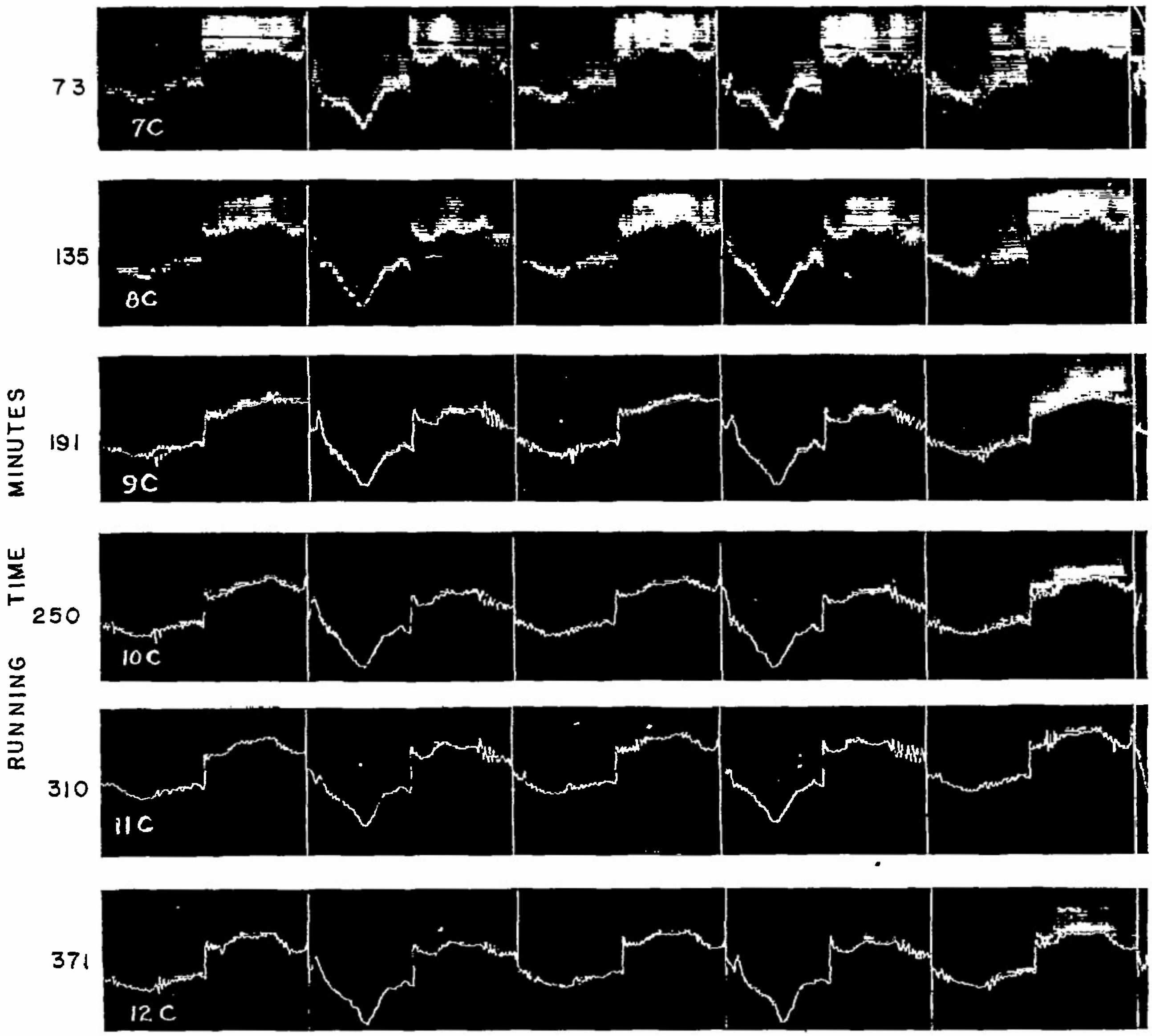


FIGURE 17.—Friction records for the run-in test with porous chrome-plated sleeve and cast-iron rings. (Records 9C to 12C have been outlined for clearness.)

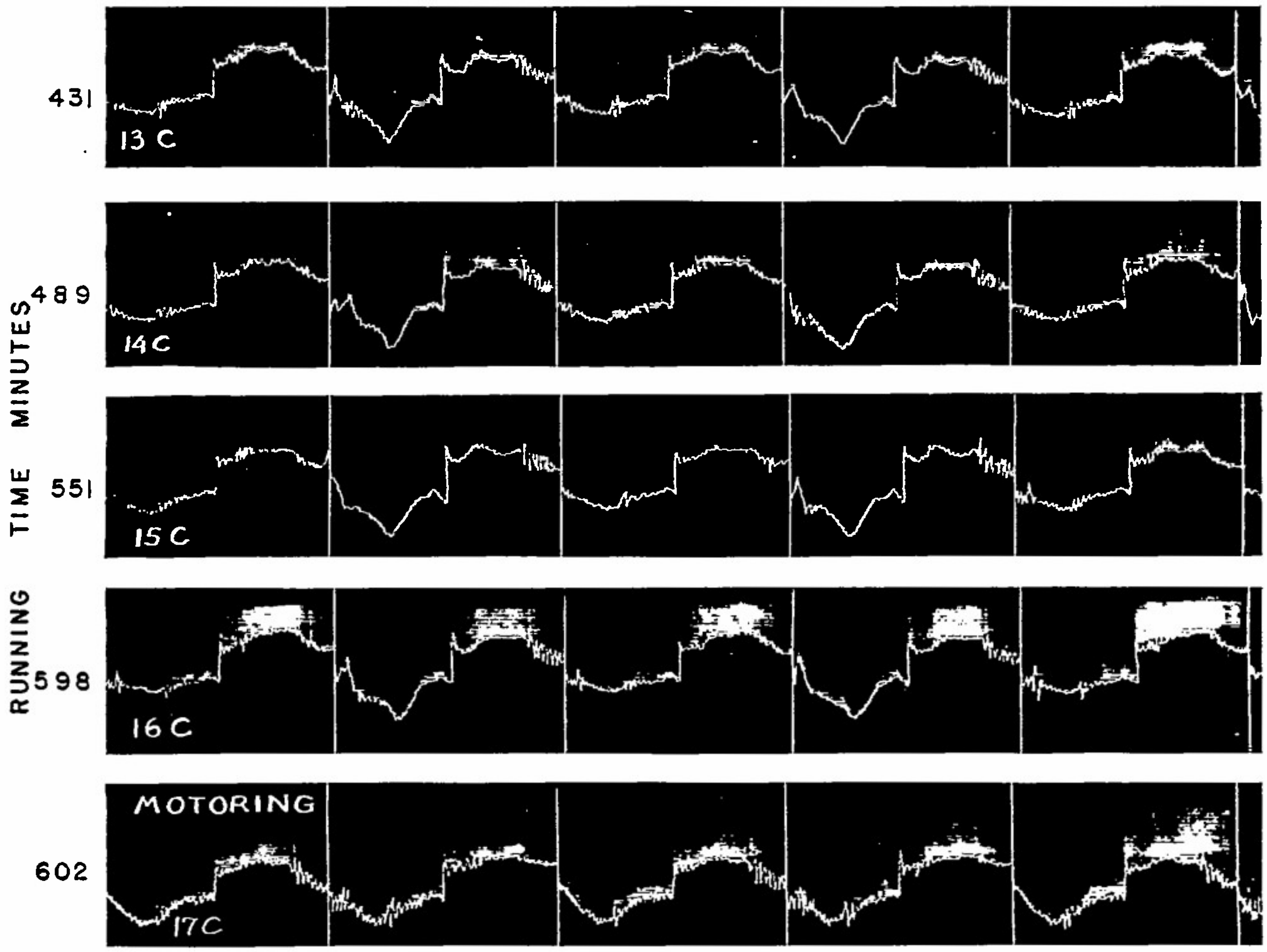


FIGURE 18.—Friction records for the run-in test with porous chrome-plated sleeve and cast-iron rings. (Records have been outlined for clearness.)

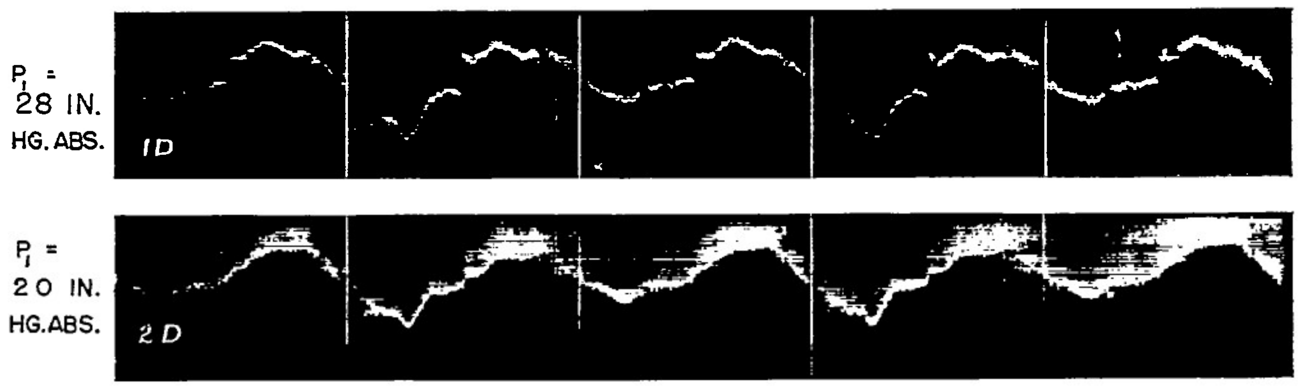


Figure 19.—Effect of manifold pressure on piston-ring friction.

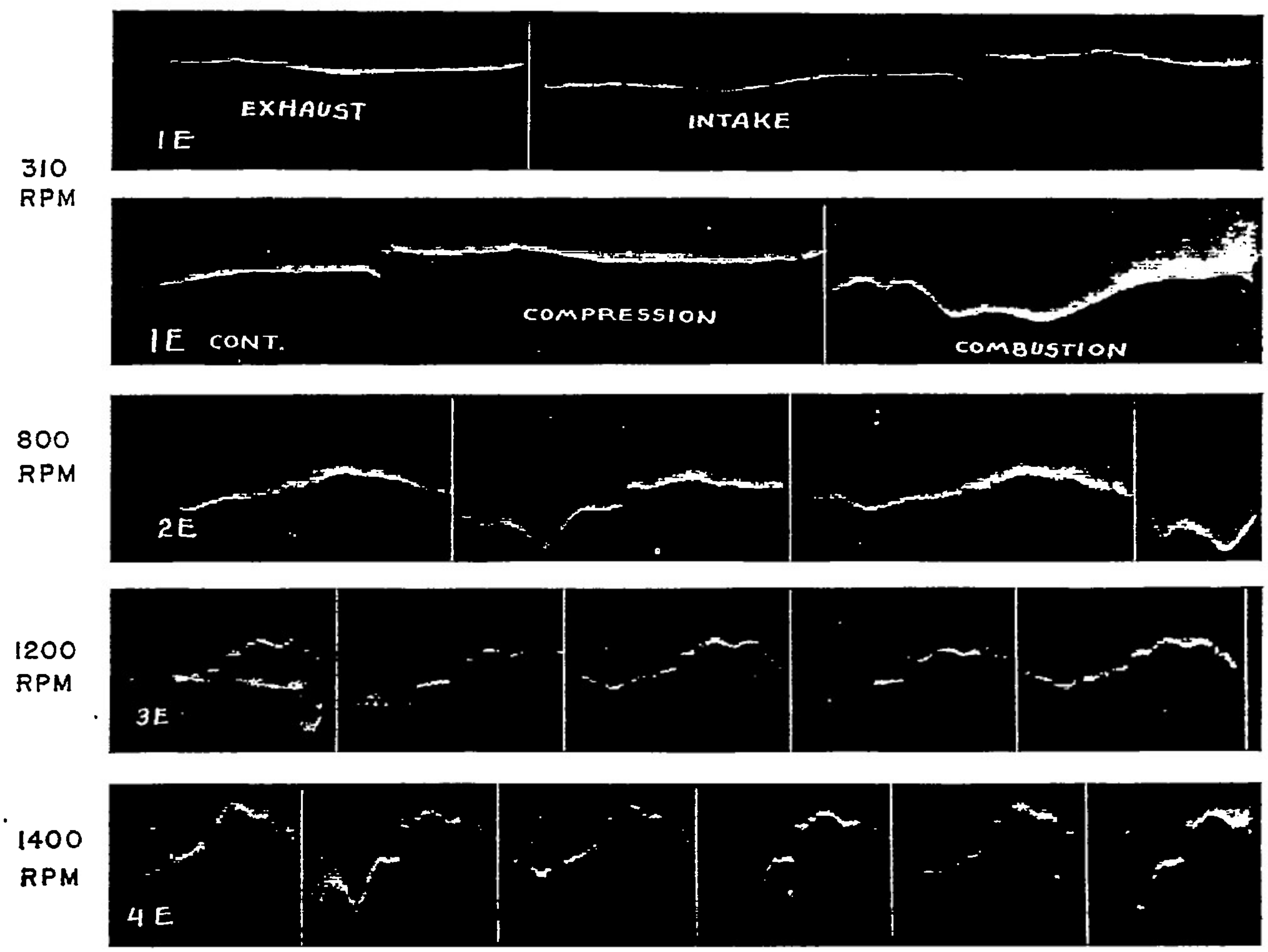


Figure 20.—Effect of engine speed on piston-ring friction.

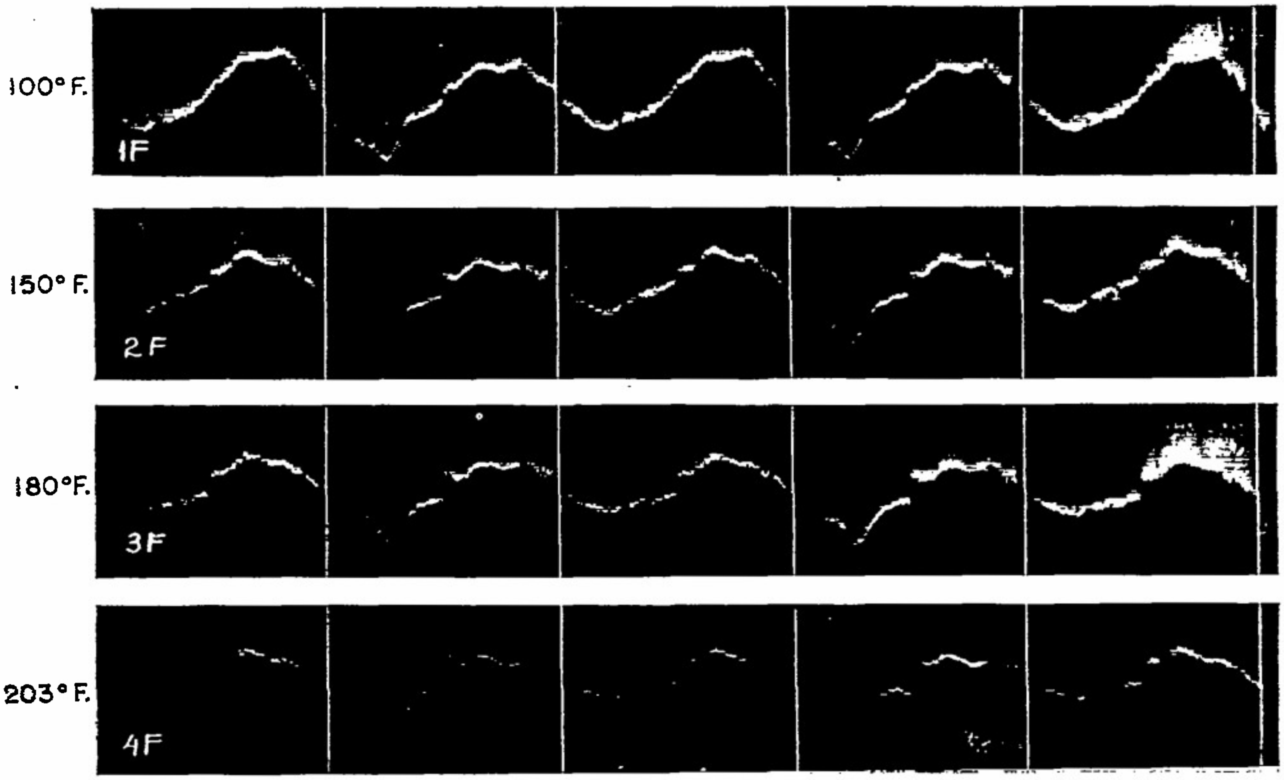


Figure 21.—Effect of cylinder jacket temperature on piston-ring friction.

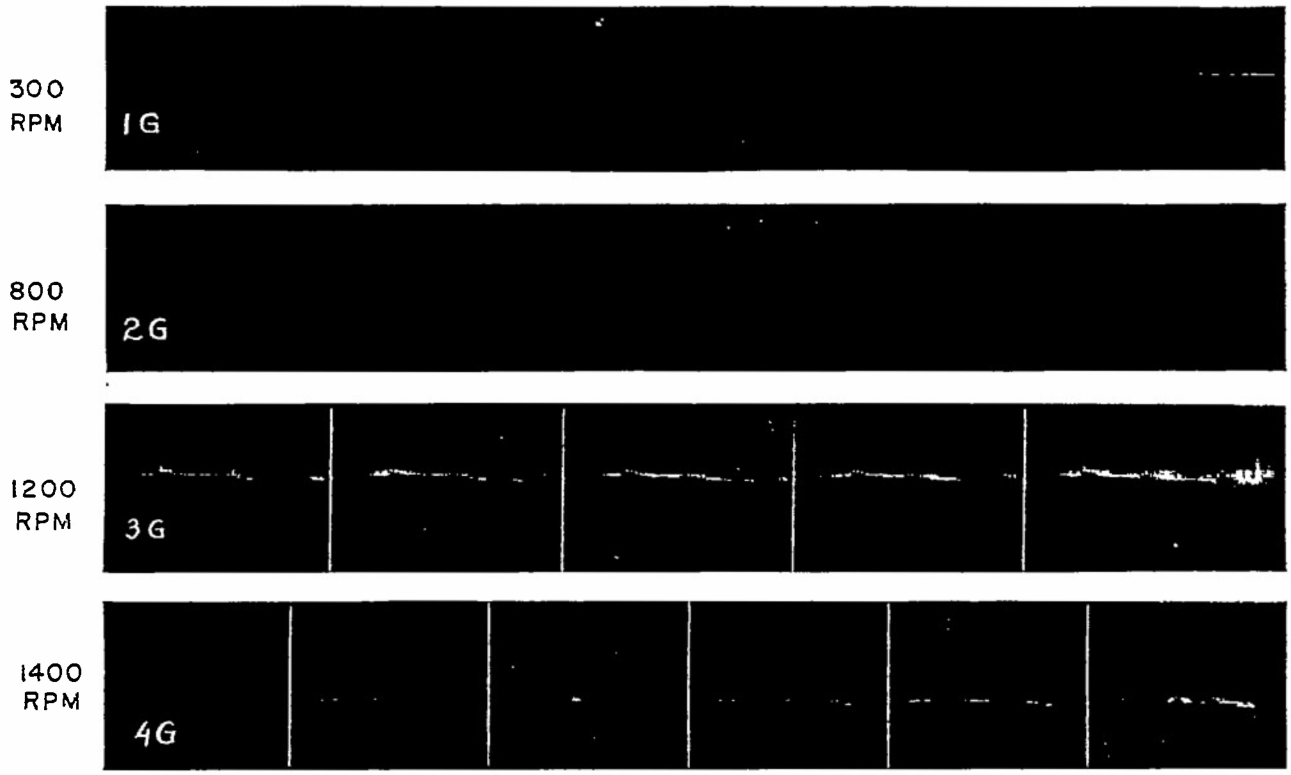


Figure 22.—Behavior of the friction-measuring apparatus as a vibration pickup. Engine motoring, piston rings, and cylinder head removed.

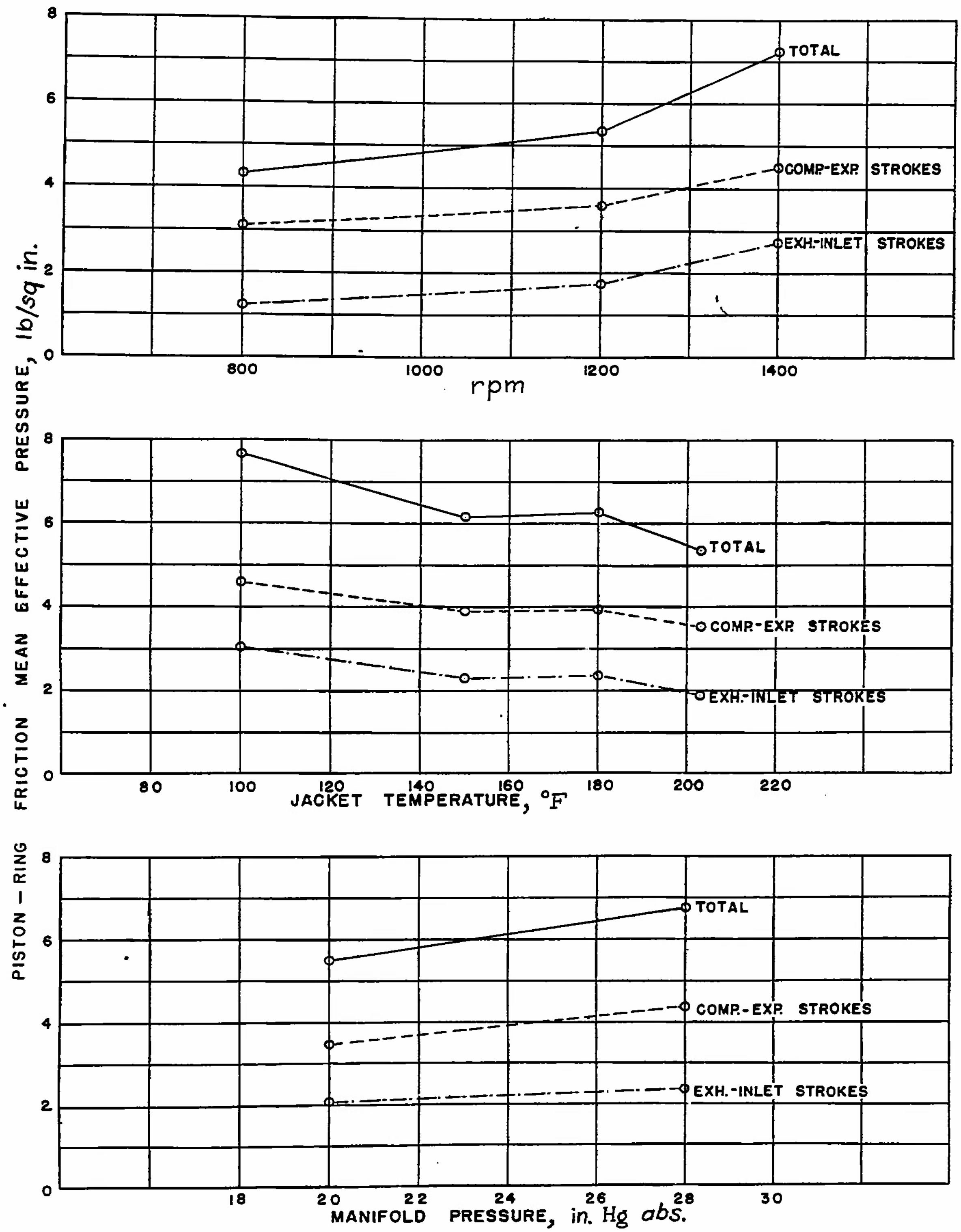
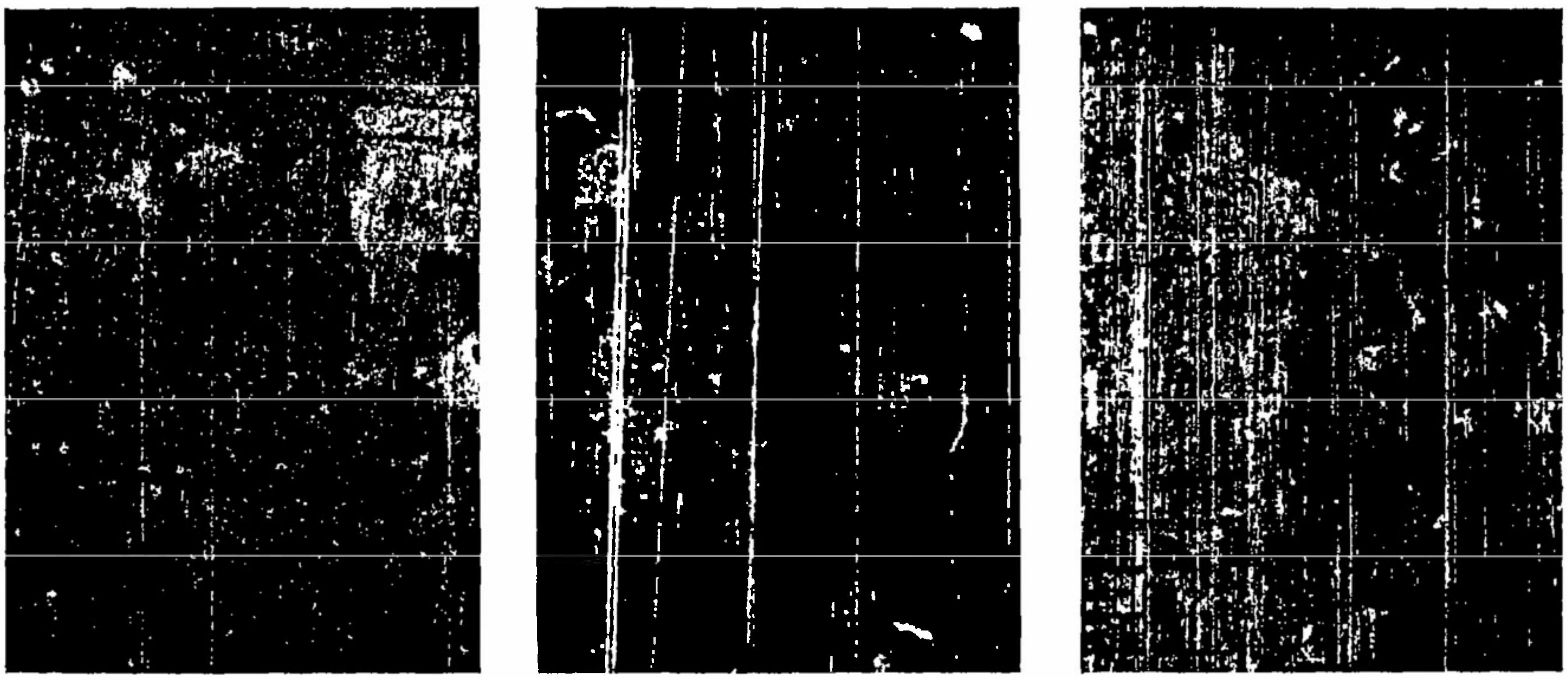


Figure 23.- The effect of engine speed, jacket temperature, and manifold pressure on piston-ring friction mean effective pressure.



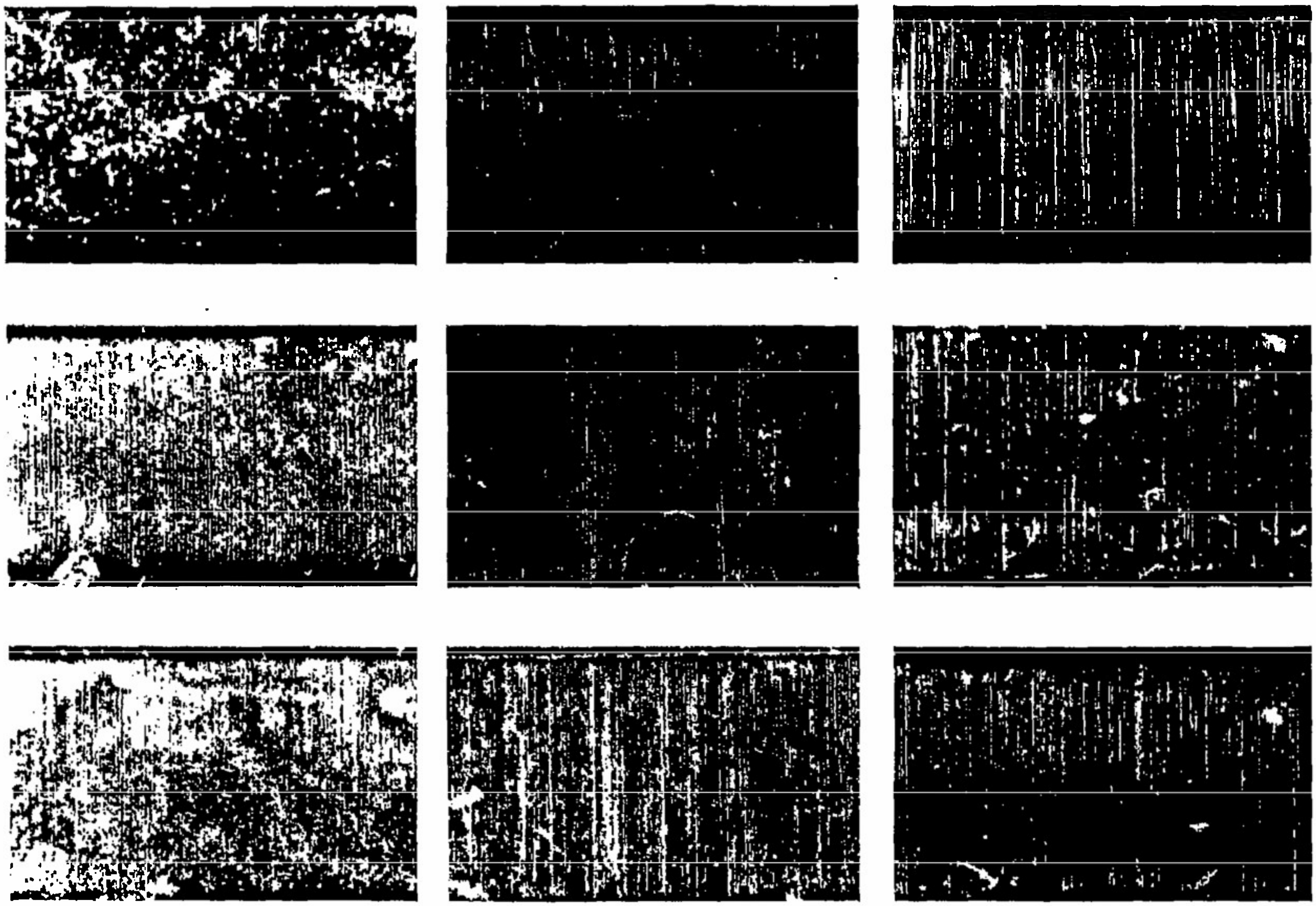
New lapped surface. After 1 hour. After 10 hours.

Figure 24.—Photomicrographs of surface replicas of SAE 4140 cylinder used in run A. (25×).

Top

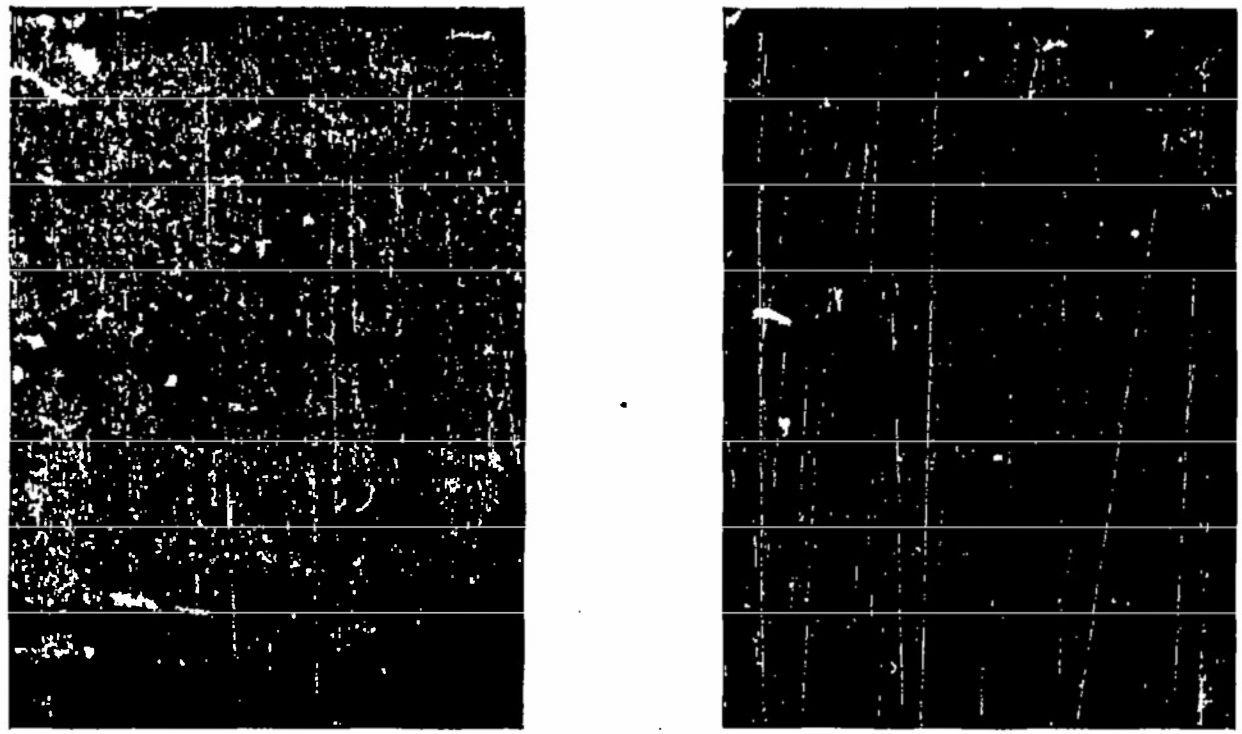
2

3



New. After 1 hour. After 10 hours.

Figure 25.—Photomicrographs of surface replicas of cast-iron rings used in run A. (25×).



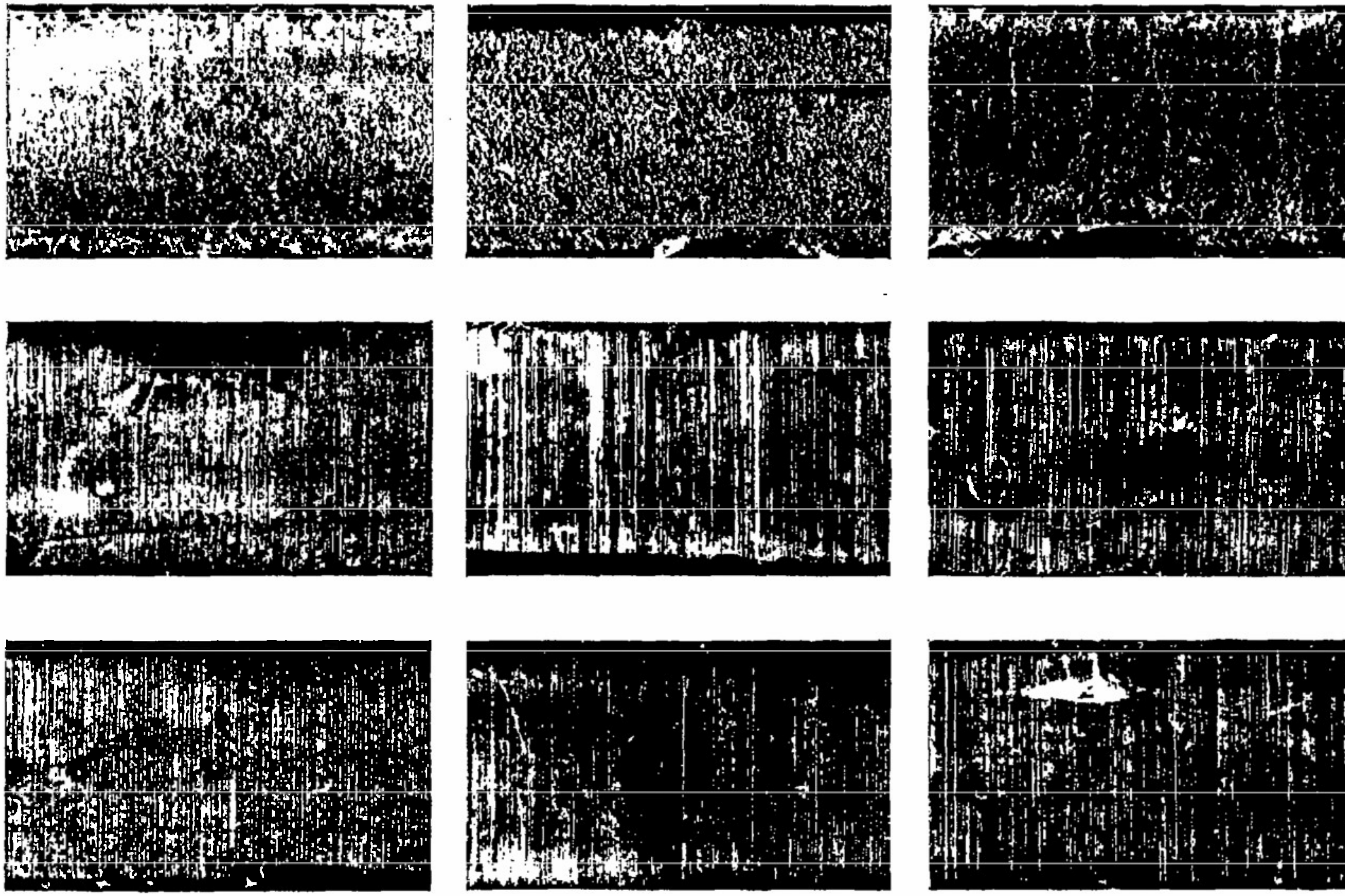
After 1 hour. After 10 hours.

Figure 26.—Photomicrographs of surface replicas of SAE 4140 cylinder used in run B. (25×).

Top

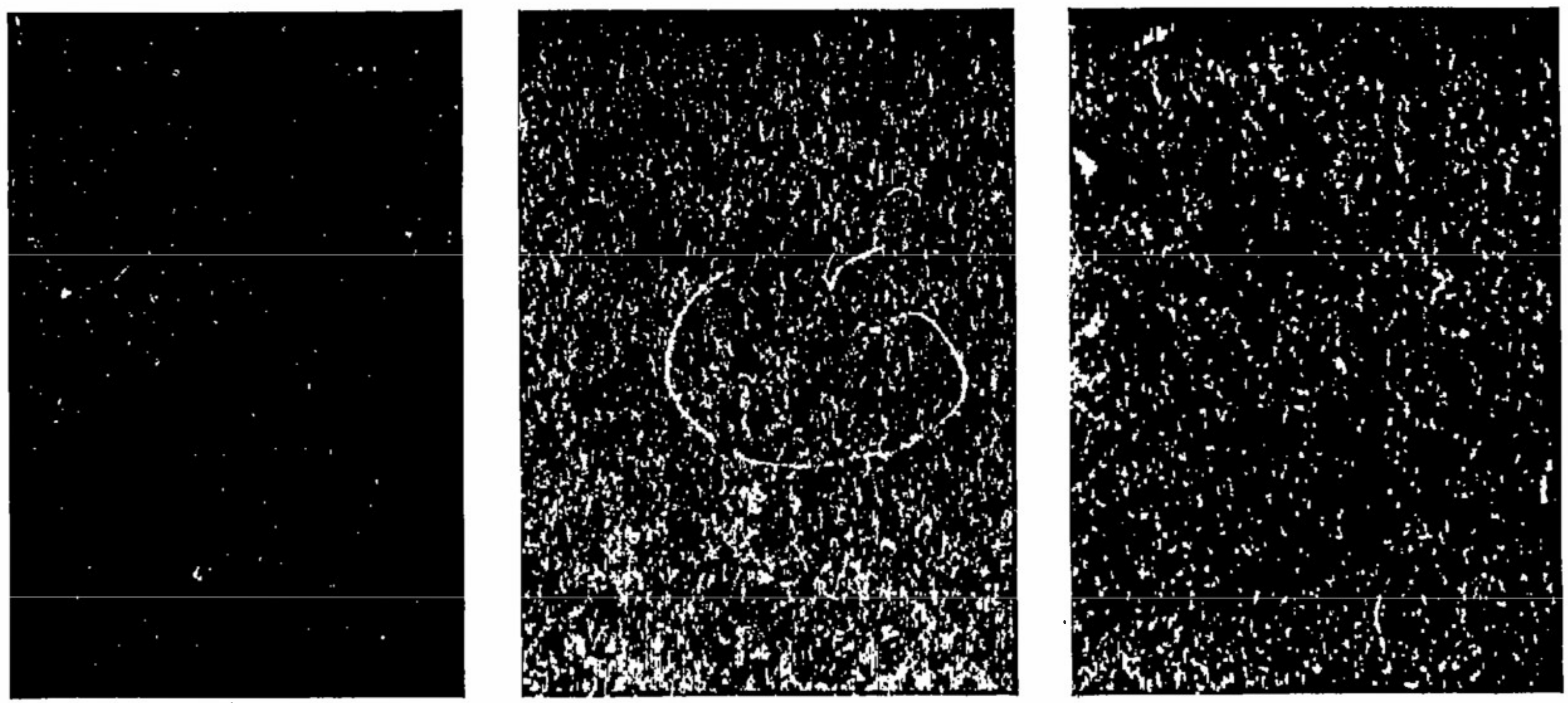
2

3



New. After 1 hour. After 10 hours.

Figure 27.—Photomicrographs of surface replicas of rings used in run B. Top ring is porous chrome-plated. (25×).



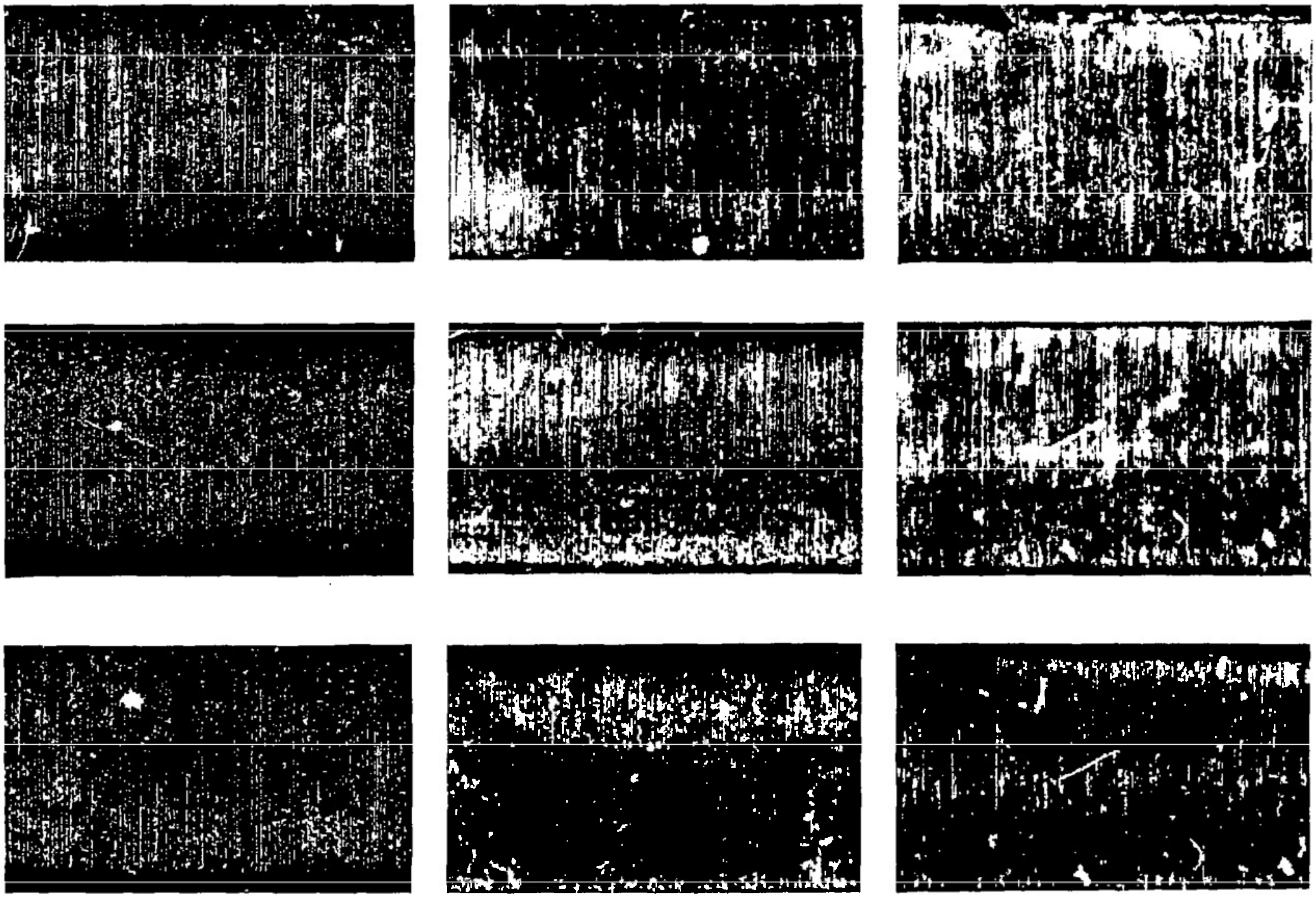
New honed surface. After 1 hour. After 10 hours.

Figure 28.—Photomicrographs of surface replicas of porous chrome-plated cylinder used in run C. (25×).

Top

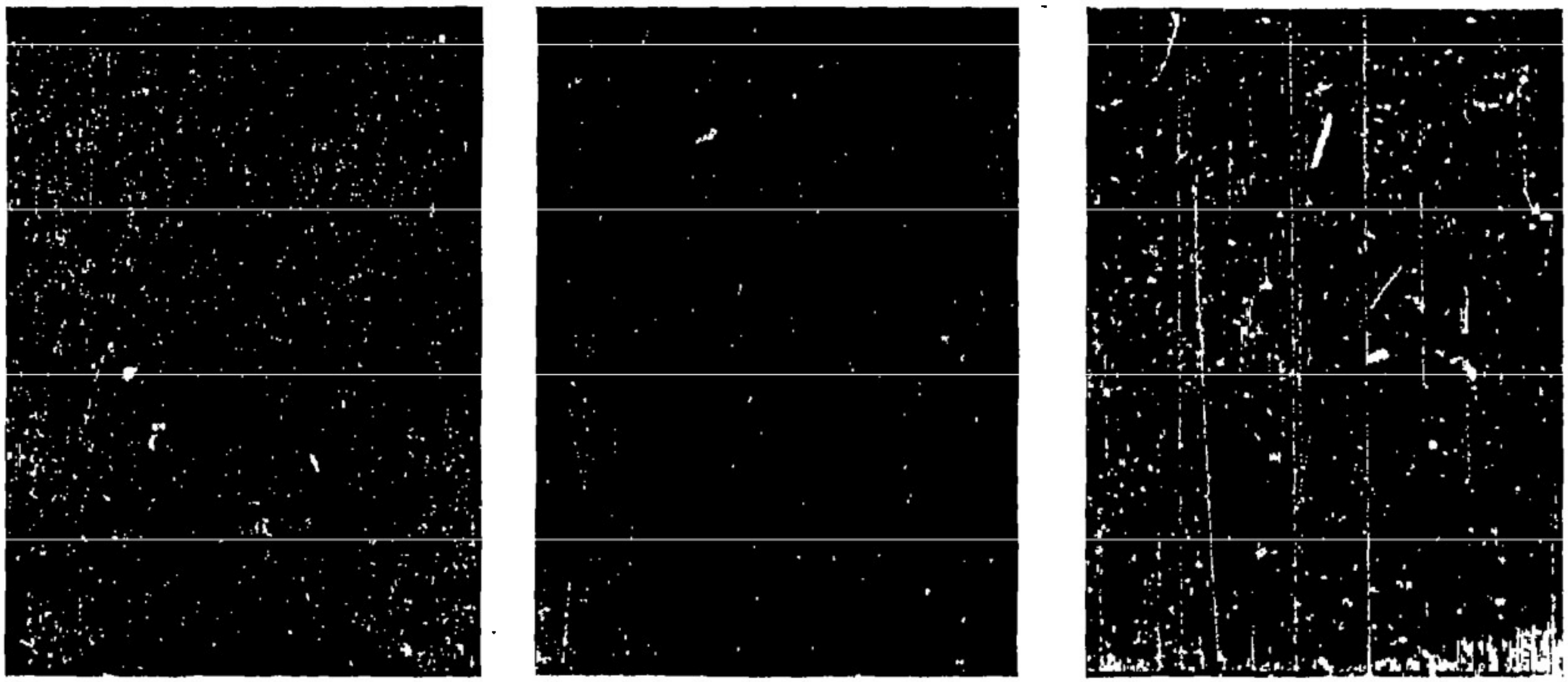
2

3



New. After 1 hour. After 10 hours.

Figure 29.—Photomicrographs of surface replicas of cast-iron rings used in run C. (25×).



New lapped surface. After 1 hour. After many hours.

Figure 30.—Photomicrographs of surface replicas of SAE 4140 cylinder used in runs D, E, and F. (25×).

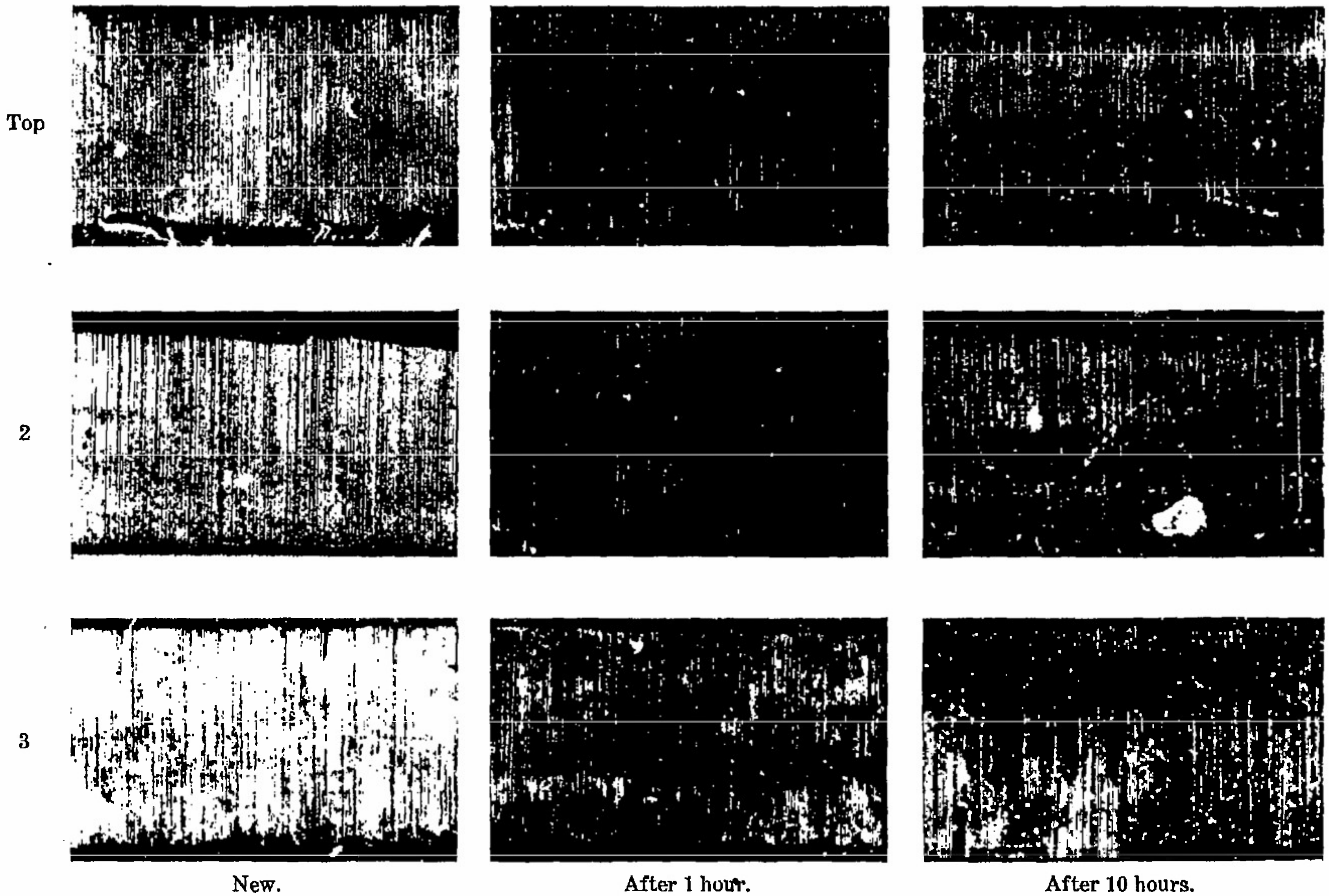


Figure 31.—Photomicrographs of surface replicas of cast-iron rings used in runs D, E, and F. (25×).

TITLE: A Study of Piston-Ring Friction	ATI- 12249
	REVISION none
AUTHOR(S): Livengood, J. C.; Wallour, Chapin	ORIG. AGENCY NO. TN-1249
ORIGINATING AGENCY: National Advisory Committee for Aeronautics, Washington, D. C.	PUBLISHING AGENCY NO.
PUBLISHED BY: (Same)	

DATE	DOC. CLASS.	COUNTRY	LANGUAGE	PAGES	ILLUSTRATIONS
Aug'47	Unclass.	U. S.	Eng.	48	photos, tables, diagrs, graphs

ABSTRACT:

Series of tests were conducted with a special engine equipped with a crosshead and elastically mounted combustion cylinder. Apparatus permitted isolation and measurement of friction forces existing between piston rings and cylinder wall. Results showed that porous chromium-plated cylinder caused slightly greater ring friction than smooth steel cylinder. Piston-ring friction increased with engine speed but decreased with increased cylinder jacket temperature.

DISTRIBUTION: Request copies of this report only from Originating Agency

DIVISION: Power Plants, Reciprocating (8) SECTION: Components (11)	SUBJECT HEADINGS: Piston rings - Wear tests (71226.5); Force - Measurement (41900)
ATI SHEET NO.: R-6-11-28	